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# DESIGN GUIDE FOR AIRCRAFT HYDRAULIC SYSTEMS AND COMPONENTS FOR USE WITH CHLOROTRIFLUOROETHYLENE NONFLAMMABLE HYDRAULIC FLUIDS

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This design guide documents the major physical properties of chlorotrifluoroethylene (CTFE) polymer-based nonflammable hydrualic fluids and the special considerations which must be observed in the design of aircraft hydraulic systems and components in order to obtain performance comparable to that obtained with petroleum-based hydraulic fluid per MIL-H-5606.

# FOREWORD

This design guide was prepared by the Boeing Military Airplane Company's Advanced Airplane Branch in Seattle, Washington, under Project 31453022, in conjunction with the evaluation of nonflammable fluids for fire resistant aircraft hydraulic systems under USAF Contract F33615-76-C-2064 which was conducted between May 1976 and May 1980. The final report for that program is documented in AFWAL-TR-80-2112 (Reference 1).

The work was administered under the direction of the Aero Propulsion Laboratory at the Air Force Wright Aeronautical Laboratories (AFWAL) with Mr. W. B. Campbell (AFWAL/POOS) as Project Engineer reporting to Mr. K. E. Binns. Mr. G. Gandee of the Propulsion Laboratory's Fire Protection Branch (AFWAL/POSH) provided the hydraulic fluid flammability values. The work was also monitored by the Materials Laboratory with Mr. C. E. Snyder (AFWAL/MLBT) and Mrs. L. Gschwender from the University of Dayton Research Institute providing some of the fluid properties, and Messrs. T. L. Graham and W. E. Berner (AFWAL/MLBT) providing elastomeric seal data.

The document was compiled by Mr. E. T. Raymond, the Boeing Program Manager, incorporating material generated by Mr. D. W. Huling, of the Advanced Airplane Branch, who conducted the fluid selection study and hydraulic seal tests; Messrs. R. L. Shick, E. C. Wagner, and W. E. Willard of the Wichita Branch, who conducted the hydraulic pump and servoactuator tests; and Mr. D. C. Sullivan of the Boeing Commercial Airplane Company Materials Technology Staff who conducted some of the fluid tests and provided consultation on questions regarding fluid and other material properties. Fluid property data was also provided by Mr. William Cassanos of the Halocarbon Products Corporation in Hackensack, New Jersey, and by Dr. T. R. Beck of the Electrochemical Technology Corporation in Seattle, Washington.

1. E. T. Raymond, D. W. Huling, and R. L. Shick, Fire Resistant Aircraft Hydraulic System, AFWAL-TR-80-2112, Boeing Military Airplane Co., Seattle, WA, (in press).

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## 1.0 PURPOSE AND SCOPE

The purpose of this design guide is to document the major physical properties of chlorotrifluoroethylene (CTFE) polymer-based nonflammable hydraulic fluids and special considerations which must be observed in the design of hydraulic systems and components intended for use with these fluids. Properties of the standard petroleum-based hydraulic fluid per specification MIL-H-5606 are also included for comparison; and, the special design considerations for the CTFE fluid are primarily those which differ with the considerations used for designing systems and components for use with MIL-H-5606 hydraulic fluid.

## 2.0 BACKGROUND INFORMATION

The Air force experiences a number of aircraft fires each year which involve the petroleum-base hydraulic fluid per MIL-H-5606 in general use in Air Force aircraft. The majority of non-combat fires involving hydraulic fluid occur on the ground or at low altitude in the wheel well and engine areas.

Previous efforts to develop a nonflammable hydraulic system have been unsuccessful due to the constraints heretofore imposed by the requirement that a new fluid must be compatible with both MIL-H-5606 fluid and present-day hydraulic systems and components. However, those constraints have been lifted; and, in a research development program undertaken to establish and verify parameters for designing fire resistant aircraft hydraulic systems, (under USAF Contract F33615-76-C-2064), a CTFE nonflammable fluid was selected as the most promising for component testing and further refinement for use in future Air Force aircraft. It is a hydrogen-free polymer with an excellent degree of nonflammability; and, most of its properties are comparable to MIL-H-5606 fluid except for its higher density and viscosity which increases system weight.

It should be noted that the fluid properties included herein, and many of the corresponding design considerations, are primarily for a specific CTFE fluid formulation: Halocarbon Products Corporation's AO-8 fluid. However, the viscosities of two other Halocarbon CTFE fluids, AO-2 fluid (which was used in the Fireproof Brake Hydraulic System Research and Development Program reported in Reference 2) and their 1.8/100 fluid (which was considered in the weight reduction study summarized in Appendix B herein) are also presented. The AO-8 fluid contains a viscosity index (V.I.) improver. The AO-2 and 1.8/100 fluids do not.

It should also be noted that the CTFE fluid is still under development and that potential problems are being addressed by AFWAL through formulation and basestock modifications.

2. S. M. Warren and J. R. Kilner, Fireproof Brake Hydraulic System, AFWAL-TR-81-2080, Boeing Military Airplane Co., Seattle, WA, September 1981.

The CTFE fluid property data was obtained from the following sources which are noted along with the values shown.

a. Fluids, Lubricants, and Elastomers Branch
   Nonmetallic Materials Division, Materials Laboratory
   Air Force Wright Aeronautical Laboratories (AFWAL/MLBT Data)

b. Fire Protection Branch
   Fuels and Lubrication Division, Aero Propulsion Laboratory
   Air Force Wright Aeronautical Laboratories (AFWAL/POSH Data)

c. The Boeing Company (Boeing Data)

d. Halocarbon Products Corporation (Halocarbon Data)

e. Electrochemical Technology Corporation (Electrochemical Technology Data)

MIL-H-5606 fluid data are also included herein for comparison. Traditional characteristics were taken from SAE Aerospace Information Report AIR 1362 (Reference 3), and special characteristics were obtained from the foregoing sources as noted herein.

3. SAE AIR 1362, Aerospace Information Report, Physical Properties of Hydraulic Fluids, Society of Automotive Engineers, Inc., Warrendale, PA, May 1975.

## 3.0 HYDRAULIC SYSTEM DESIGN CONSIDERATIONS

This section provides information relating to the design of hydraulic systems. Information relating to the design of hydraulic components is provided in Section 4.0.

The overall impact of the CTFE fluid upon a complete aircraft hydraulic system may be described in terms of the predicted changes in operating performance, increases in weight and cost, and potential changes in reliability, maintainability, and safety. However, it is assumed that no major reductions in performance can be tolerated and that system components and tubing runs will be designed to provide flow at rates necessary to meet specified operational requirements.

### 3.1 System Arrangement

Normal considerations can be observed in arranging hydraulic systems for use with the CTFE fluid.

### 3.2 Component Location and Tube Routing

Due to the CTFE fluid's nonflammability, there is more freedom of choice because there is:

a. No need to avoid heat sources and other sources of combustion.
b. No need for firewalls or shrouds for hydraulic lines and components.
c. No need to preclude tubing from the cabin except that tubing should not be located where a leak could scald personnel or damage equipment or cargo.

### 3.3 Tube Sizing

Hydraulic fluid transmission lines fall into three general classifications: pressure lines, return lines, and pump suction lines. Due to the CTFE fluid's higher density, and the higher low-temperature absolute viscosity of the AO-8 fluid, tube sizes somewhat larger than those used for MIL-H-5606 fluid systems will be required for the AO-8 fluid. However, other CTFE fluid formulations, such as the AO-2 and the 1.8/100 fluid, which have lower viscosities than the AO-8 fluid, are available; and, smaller tube sizes could be used. See Appendix B for an example of the weight swing which could be realized through the use of the 1.8/100 fluid on a transport aircraft.

### 3.3.1 System Pressure and Return Lines

The choice of tubing sizes for pressure and return lines is usually based on trade studies which balance energy loss against cost and weight. Large diameter tubing will conduct the fluid with lower pressure loss than smaller sizes, but will cost and weigh more.

Basically, tubing must be large enough so that, at all design conditions, pressure losses will not prevent all actuators from meeting their load and rate requirements. Secondly, the sizes must be large enough to prevent harmful dynamic pressures due to high fluid velocities. MIL-H-5440 specifies that "peak pressure resulting from any phase of the system operation shall not exceed 135 percent of the main system, subsystem, or return system pressure when measured with electronic equipment, or equivalent." The following discussions of those factors include the relevant equations for quantitative calculations.

#### a. Pressure-Loss Analysis

The typical hydraulic transmission system operating pressure cycle is illustrated graphically in Figure 1 which depicts the following four main phrases of the cycle:

(1) The rise in pressure across the system pump.

(2) The pressure loss in the system pressure lines.

(3) The differential pressure available for actuating loads.

(4) The pressure loss in system return lines.

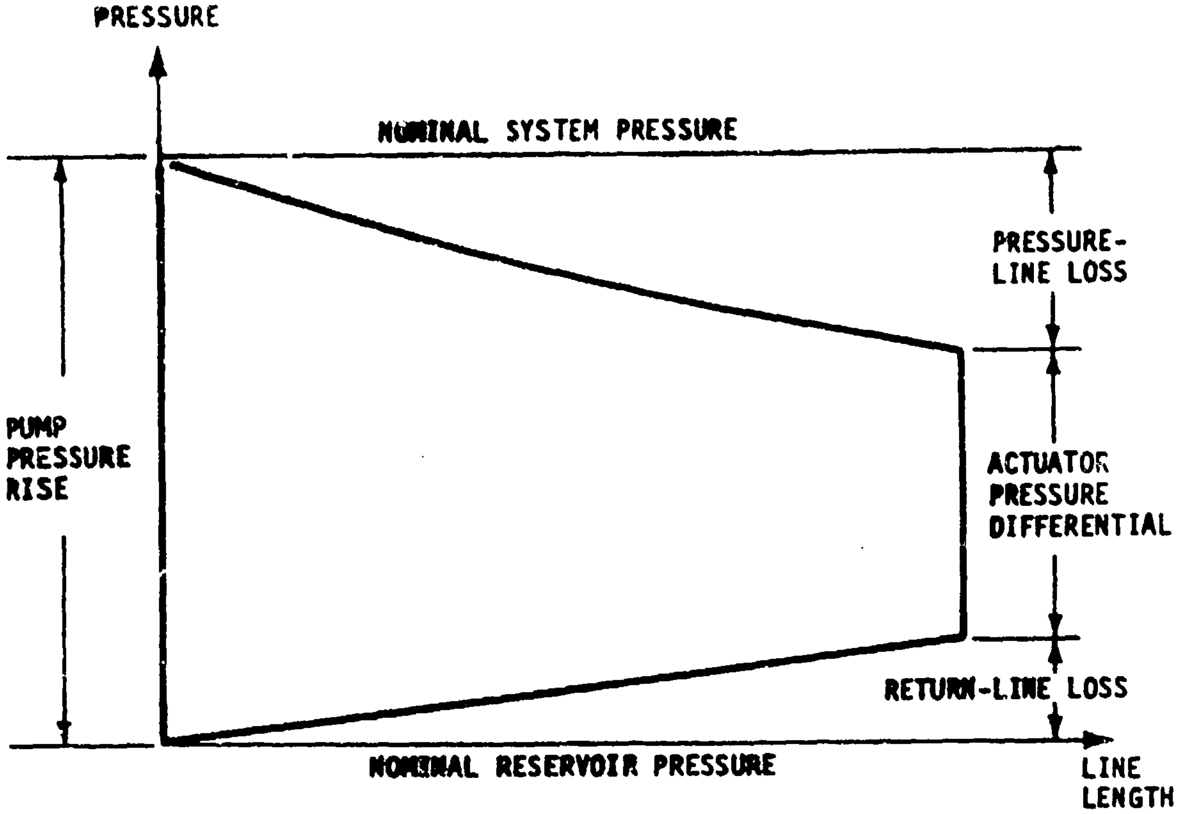


Figure 1 Typical hydraulic system pressure cycle

When the actuator loads, rates, and aircraft operating temperature requirements have been defined, the required hydraulic fluid flow rates and allowable pressure losses at various fluid operating temperatures can be established. From those requirements, and the necessary tube lengths for installation in the air vehicle, the required tube sizes can be calculated. This is generally done utilizing equations derived from the Darcy-Weisbach formula for lost head in round pipes, namely:

$$h_f = f \frac{L}{D} \frac{V^2}{2g} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

which was presented first in a somewhat more general form by Antoine Chezy in 1775. That formula has been reduced to the following form in order to utilize the generally used engineering units noted below:

$$\Delta P = 0.0135 \frac{fLsQ^2}{D^5} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

where:

$\Delta P$ = Pressure loss, pounds per square inch (psi)

f = Friction factor, dimensionless

L = Length of tube, feet (ft)

Q = Fluid flow rate, US gallons per minute (gpm)

D = Tube inside diameter, inches (in)

s = Fluid specific gravity, dimensionless, or fluid density, grams per cubic centimeter (g/cm$^3$)

The friction factor (f) varies as a function of Reynolds number as shown in Figure 2 which can be found in several textbooks and other reference documents including the SAE Aerospace Recommended Practice ARP 24B (Reference 4). As can be seen in Figure 2, the relationship between friction factor and Reynolds number follows the following equations for laminar flow and turbulent flow respectively:

For laminar flow: $$f = \frac{64}{N_R} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

For turbulent flow: $$f = \frac{0.316}{N_R^{0.25}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

As shown in ARP 24B, Reynolds number ($N_R$) is also a dimensionless factor which can be expressed in either of the following formulae:

$$N_R = \frac{\rho VD}{\mu} \quad \ldots\ldots\ldots \quad (5) \qquad \text{or} \qquad N_R = \frac{VD}{\nu} \quad \ldots\ldots\ldots\ldots \quad (6)$$

4. SAE ARP 24B, Aerospace Recommended Practice, Determination of Hydraulic Pressure Drop, Society of Automotive Engineers, Inc., Warrendale, PA, 1-31-68.

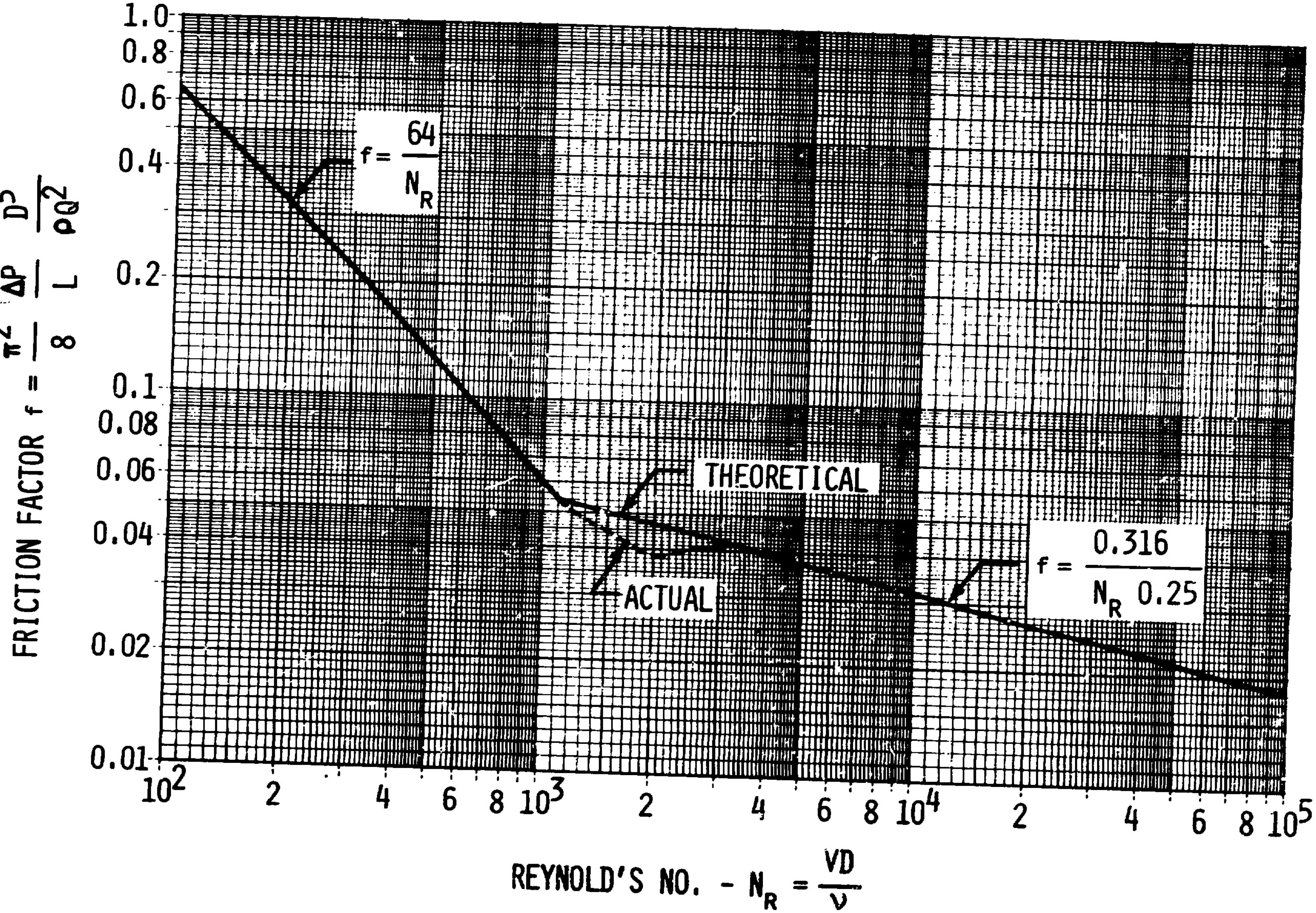


Figure 2 Friction factor f as a function of Reynolds number $N_R$ for smooth tubes

Substituting rate of fluid flow (Q) for fluid velocity (V), and specific gravity (s) for fluid mass density ($\rho$), results in the following forms:

$$N_R = 3160 \frac{sQ}{\mu D} \quad \ldots\ldots (7) \qquad \text{or} \qquad N_R = 3160 \frac{Q}{\nu D} \quad \ldots\ldots\ldots (8)$$

where:

s = Fluid specific gravity, dimensionless, or
fluid density, grams per cubic centimeter ($g/cm^3$)

Q = Fluid flow rate, US gallons per minute (gpm)

D = Tube inside diameter, inches (in)

$\mu$ = Fluid absolute viscosity, centipoises (cp)

$\nu$ = Fluid kinematic viscosity, centistokes (cs)

In general, for straight tubing, laminar flow is predominant for values of $N_R$ below about 1,400, and becomes fully turbulent above about 3,600 (see Figure 2). When the flow is disturbed by the presence of bends and fittings, a turbulent condition is found to prevail down to $N_R$ of 1,000 or less.

It is generally recognized that, under normal operating temperatures, high flow demands in aircraft hydraulic systems generally result in turbulent flow. However, tubing sizes are almost always determined by the requirement to keep pressure losses within established allowable limits under low-temperature conditions before the fluid has warmed up to its normal operating temperature. Under such conditions, flow is almost always laminar.

The friction factors for laminar flow and turbulent flow can be calculated from the following equations which were derived by substituting Eq.(7) into Eq. (3) and Eq.(4) respectively:

For laminar flow: $$f = 0.02025 \frac{\mu D}{sQ} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (9)$$

For turbulent flow: $$f = 0.042 \left(\frac{\mu D}{sQ}\right)^{0.25} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (10)$$

The laminar-flow and turbulent-flow pressure losses in tubing runs can be calculated from the following equations which were derived by substituting the foregoing friction factors into Eq (2).

For laminar flow: $$\Delta P = 0.000{,}273 \frac{\mu QL}{D^4} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (11)$$

For turbulent flow: $$\Delta P = 0.000{,}569 \frac{\mu^{0.25} s^{0.75} Q^{1.75} L}{D^{4.75}} \quad \ldots\ldots \quad (12)$$

The foregoing equations can also be expressed in the following forms which utilize the more readily available kinematic viscosity values:

For laminar flow: $$f = 0.02025 \frac{\nu D}{Q} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (13)$$

$$\Delta P = 0.000{,}273 \frac{\nu sQL}{D^4} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (14)$$

For turbulent flow: $$f = 0.042 \left(\frac{\nu D}{Q}\right)^{0.25} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (15)$$

$$\Delta P = 0.000{,}569 \frac{\nu^{0.25} s Q^{1.75} L}{D^{4.75}} \quad \ldots\ldots\ldots \quad (16)$$

When calculating pressure losses for pressure lines, it is extremely important that the correct fluid viscosity values be used. The following tabulation of kinematic viscosities at -50F, 0F, and +50F fluid temperatures show that, for MIL-H-5606 fluid, the viscosity at 3,000 psi is approximately double that at atmospheric pressure; and, that the multiplication factor increases as fluid temperature decreases. At higher pressures, the multiplication factor is significantly higher.

| MIL-H-5606 Fluid Pressure (psi) | atmos. | 2,000 | 4,000 | 6,000 | 8,000 | 10,000 |
|---|---|---|---|---|---|---|
| Kinematic Viscosity at -50F (cs) | 740 | 1,200 | 1,900 | 3,200 | 5,000 | 8,000 |
| Kinematic Viscosity at 0F (cs) | 100 | 140 | 200 | 285 | 410 | 580 |
| Kinematic Viscosity at +50F (cs) | 30 | 42 | 56 | 76 | 100 | 140 |

The foregoing values were taken from Figure 13 in AIR 1362 (Reference 3). Similar data is needed for CTFE fluid before accurate pressure-loss analyses can be made for CTFE fluid systems. However, if in the meantime it is assumed that the pressure multiplication factors for CTFE fluids will be similar to those for MIL-H-5606 fluid systems, the following equations can be used to estimate the change in tubing diameters which would be required to convert an existing MIL-H-5606 fluid system for use with a CTFE fluid.

For laminar flow: $$\frac{D_2}{D_1} = \left(\frac{\mu_2}{\mu_1}\right)^{0.25} \quad \ldots\ldots\ldots\ldots\ldots (17)$$

$$\frac{D_2}{D_1} = \left(\frac{\nu_2 s_2}{\nu_1 s_1}\right)^{0.25} \quad \ldots\ldots\ldots\ldots\ldots (18)$$

For turbulent flow: $$\frac{D_2}{D_1} = \left(\frac{s_2}{s_1}\right)^{3/19}\left(\frac{\mu_2}{\mu_1}\right)^{1/19} \quad \ldots\ldots\ldots\ldots (19)$$

$$\frac{D_2}{D_1} = \left(\frac{s_2}{s_1}\right)^{4/19}\left(\frac{\nu_2}{\nu_1}\right)^{1/19} \quad \ldots\ldots\ldots\ldots (20)$$

## b. Velocity Limits and Pressure Peaks

As previously noted, MIL-H-5440 specifies that "peak pressure resulting from any phase of the system operation shall not exceed 135 percent of the main system, subsystem, or return system pressure when measured with electronic equipment or equivalent." The velocity of the hydraulic fluid flowing through system tubing direcly affects the magnitude of peak pressure surges when an abrupt valve closure is initiated under a high-flow condition. The magnitude of the pressure rise above the normal system pressure can be calculated from the following formula:

$$\Delta P = 12V\sqrt{\beta\rho} \qquad (21)$$

where:

$\Delta P$ = Pressure rise, pounds per square inch (psi)
$V$ = Original fluid velocity, feet per second (fps)
$\beta$ = Fluid bulk modulus, pounds per square inch (psi)
$\rho$ = Fluid mass density, $lb\text{-}sec^2/in^4$

As seen in Appendix B, typical limiting velocities for MIL-H-5606 fluid and AO-8 CTFE fluid in 3,000-psi systems, as required to keep the pressure rise within 35 percent of system pressure (1,050 psi), will be on the order of 25 and 20 feet per second respectively depending upon the fluid operating temperature and the assumed fluid system compliance. It should be noted that typical values of fluid system compliance are somewhat smaller than the fluid bulk modulus. Laboratory measurements of fluid bulk modulus are generally always made with all entrained air and other gasses carefully removed, whereas in actual fluid systems the compliance is reduced by the effect of such entrained gasses and by the elasticity of the tubing and hoses.

It may be noted that these values are greater than the historical fluid velocity limitation of 15 feet per second which was specified in the original issue of MIL-H-5440 and in all subsequent revisions through Revision D. However, in MIL-H-5440E and in subsequent issues, the reference to the 15 fps limitation was replaced with the following requirement:

"Fluid velocity limitations - Tubing size and maximum fluid velocity for each system shall be determined considering, but not limited to, the following:

(a) Allowable pressure drop at minimum required operating temperatures.
(b) Pressure surges caused by high fluid velocity and fast response valves.
(c) Back pressure in return lines, as it may affect brakes and pump case drain lines.
(d) Pump inlet pressure, as affected by long suction lines, and a high response rate variable pump. Consideration should be given to both pressure surges and cavitation."

### 3.3.2 Pump Suction Lines

The size of the pump suction lines must be adequate to ensure flow to the pump upon startup, and adequate pressure at the pump inlet port to preclude cavitation damage of the pump during all expected flow demands. To move hydraulic fluid from the system reservoir to the pump, sufficient reservoir pressure must be provided to both overcome the steady-state pressure losses in the pump suction line and also to accelerate the column of fluid in the suction line. Cavitation will occur if the system designer fails to match the flow response of the inlet system to the response required by the discharge flow demands from the pump.

The steady-state pressure losses can be calculated with the same formulae used for pressure and return lines making sure that the atmospheric-pressure viscosity values are used. The additional pressure required to ensure adequate response can be determined from the basic equation for acceleration force ($F = ma$) as follows:

$$F = \rho AL\left(\frac{dV}{dt}\right) = \rho AL\left(\frac{dQ/A}{dt}\right) = \rho L\left(\frac{dQ}{dt}\right)$$

The equivalent flow-response pressure requirement can be expressed as follows:

$$P_{resp} = \frac{F}{A} = \frac{\rho L}{A}\left(\frac{dQ}{dt}\right) = \frac{\rho L}{\pi D^2/4}\left(\frac{dQ}{dt}\right) \quad \text{(22)}$$

When converted to a form which utilizes the commonly used engineering units for fluid density (grams per cubic centimeter), for line length (feet), and for flow rate (gallons per minute), the foregoing formula appears as follows:

$$P_{resp} = 0.0055\,\frac{\rho L}{D^2}\left(\frac{dQ}{dt}\right) \quad \text{(23)}$$

where:

$P_{resp}$ = Flow-response pressure requirement (psi)

$\frac{dQ}{dt}$ = Flow-response requirement (gpm/s)

L = Line length (ft)

D = Tube inside diameter (in)

The required pump suction line tubing diameter can be calculated by utilizing the following equation for the total suction line pressure requirement. However, it should be noted that at least two operating conditions need to be examined to ensure sufficient pressure at the pump inlet port to "prime" the pump at the minimum operating temperature (when the fluid viscosity is high), and to ensure that the flow through the suction line responds to pump output flow demands sufficiently to preclude damaging cavitation of the pump.

$$P_{resv} = P_{crit} + P_{resp} + \Delta P_{comp} + \Delta P_{line} \quad \ldots \ldots \ldots \ldots \ldots \quad (2A)$$

where:

$P_{resv}$ = Reservoir pressure

$P_{crit}$ = Pump critical inlet pressure (see Para, 4.2.b)

$P_{resp}$ = Flow-response pressure requirement

$\Delta P_{comp}$ = Pressure loss in suction-line components, ie: shutoff valve, check valve, self-sealing disconnect coupling, etc.

$\Delta P_{line}$ = Pressure loss in suction-line tubing and hose

## 3.4 Impact on System Weight

The impact of the CTFE fluid on the weight of hydraulic system components, as compared to components designed for use with fluid per MIL-H-5606, is primarily due to the contained fluid rather than the housings. However, significant increases in the weight of hydraulic lines can also be expected with the use of the AO-8 fluid due to the increased sizes necessary to maintain pressure losses within desired limits. The increased line sizes will increase the volume of fluid in the system; and, this, in turn, could require larger reservoirs and possibly larger heat exchangers.

Balancing those weight increases to some extent, will be the weight decreases which can be realized by eliminating pump suction-line shutoff valves and other fire safety provisions such as firewalls and shrouds installed to isolate hydraulic fluid from ignition sources.

To get a better view of what the line diameter changes mean in weight terms and the effect of the flow design temperature upon the CTFE-system to MIL-H-5606-system weight ratio, a weight ratio equation was developed for use with the diameter ratios determined through the use of Eq. (17) through Eq. (20).

Starting with the weight relation $W_{TOTAL} = W_{TUBE} + W_{FLUID}$, the following wet tube weight ratio can be used for comparison of the CTFE-system tubing weight with the MIL-H-5606-system tubing weight.

$$\frac{W_2}{W_1} = \frac{D_2^{\,2}}{D_1^{\,2}} \left( \frac{\rho_2 + 4\rho_t \left[ \frac{t_2}{D_2} + \frac{t_2^{\,2}}{D_2^{\,2}} \right]}{\rho_1 + 4\rho_t \left[ \frac{t_1}{D_1} + \frac{t_1^{\,2}}{D_1^{\,2}} \right]} \right) \quad \ldots\ldots\ldots\ldots\ldots \quad (25)$$

where:

$W$ = weight, wet tube
$\rho_t$ = density, tube material
$t$ = thickness, tube wall
Subscript 1 = reference fluid parameters (MIL-H-5606)
Subscript 2 = parameter for fluid of interest (CTFE)

Weight comparisons for a typical cargo aircraft hydraulic system are shown in Appendix A. The 1987 lb weight penalty estimated for incorporating the AO-8 CTFE fluid in lieu of MIL-H-5606 fluid is approximately 57% of the original hydraulic power and distribution system weight, and 28% of the weight of the overall hydraulic system including the actuators. For smaller aircraft such as fighters, close support aircraft, and helicopters, where the distribution tubing runs are shorter, a smaller weight penalty would be expected.

A number of methods for reducing the weight penalty can be considered. One of these is to use a lower viscosity version of the CTFE fluid in order to reduce tubing sizes and fluid volume. As shown in Appendix B, the weight penalty for hydraulic tubing runs, including the fluid contained therein plus the attachment clamps and end fittings, designed for use with the Halocarbon AO-8 CTFE fluid could be reduced some 57% with the smaller sizes allowed by the use of the lower-viscosity Halocarbon 1.8/100 CTFE fluid. However, this would require hydraulic pumps and motors which can operate without lubrication failure with the reduced viscosity at their maximum operating temperatures, ie: 0.94 centipoise at 275F compared with 2.6 cp for MIL-H-5606 fluid.

Other means to reduce fluid volume involve reducing fluid flow, such as through the use of higher system operating pressures or the use of load-adaptive actuation systems, in order to reduce tube sizes; or reducing tubing length through the use of integrated actuator packages or satellite hydraulic systems located at the remote points of usage around an aircraft. However, none of the aforementioned techniques is unique to the CTFE fluid. They could also be used to reduce the weight of a hydrocarbon-base hydraulic fluid system.

Another way to reduce the weight impact of the CTFE fluid is to use it only in those portions of an overall system which are proximate to ignition sources such as engines and wheel brakes. One weight-effective approach is to use it only in the wheel brake systems. Air Force experience indicates that approximately two-thirds of their aircraft hydraulic fires occur in the wheel well areas due to fluid leaking onto hot wheel brake assemblies. Use of the CTFE fluid only in the brake systems would provide a significant reduction in hydraulic fluid fires for only a relatively small weight penalty.

Tests of a two-fluid brake hydraulic system conducted on the Fireproof Brake Hydraulic System R&D Program indicated that the concept is feasible. In that program, in which a KC-135 aircraft brake system was tested, Halocarbon AO-2 CTFE fluid was used in the brakes and brake lines downstream of a modified KC-135 brake deboost valve and MIL-H-5606 fluid upstream. The basic operation and control characteristics of the brake system were not affected by the two-fluid configuration; and, the hardware modifications had virtually no effect on system performance. As noted in the final report for that program, Reference 2, the increased density of the CTFE fluid did affect the dynamic response of the brake hydraulic system which resulted in an indicated increase in aircraft stopping distance over that obtained with the original MIL-H-5606 fluid system. However, analysis indicated that the performance lost by changing to the CTFE fluid could be regained by increasing the hydraulic line sizes, by using hard tubing rather than hoses, or by retuning the antiskid control box.

## 4.0 HYDRAULIC COMPONENT DESIGN CONSIDERATIONS

### 4.1 Materials and Standards

#### 4.1.1 Component Materials

Most of the metals and plastic materials currently used with MIL-H-5606 fluid systems can be used with CTFE fluid except that, until the fluid includes an acceptable corrosion inhibitor and/or another additive which will prevent the loss of the fluid's protective film when parts are exposed to an air atmosphere, corrosion resistant materials should be used wherever practicable. Component material compatibility data are given in Section 6.1. The following materials should be used with caution, and thoroughly evaluated by test to determine their adequacy for the intended application:

Carbon (some bonding materials may be incompatible)
Copper and copper bearing alloys

#### 4.1.2 Seals and Packings

As noted in Section 6.2, the Firestone phosphonitrilic fluoroelastomer (PNF) compound 280-C01R, with Shore hardness of 80 durometer minimum, is the best elastomer found to date for O-rings and other elastomeric seals intended for use in CTFE fluid. The material is produced by the Firestone Tire & Rubber Company, 1200 Firestone Parkway, Akron, Ohio 44317; and, can be molded by the normal O-ring suppliers in standard molds. Seal gasket plates with molded PNF elastomer can be obtained from the Parker Seal Company, 10567 Jefferson Boulevard, Culver City, California 90230.

In the event that PNF seals are unavailable, the following materials may be substituted (in the order of priority listed). However, before these materials can be considered acceptable for use in flight articles with CTFE fluid, they should be tested to determine their adequacy for the intended application.

Ethylene propylene rubber (EPR)
Hydrofluorocarbon but not for low temperature
Chlorinated polyethylene (CPE) but not below 0°F

Seal backup rings, uncut, and slipper seals made from filled polytetrafluoroethylene (PTFE) are acceptable. C. E. Conover & Co.'s Revonoc 18158 and Royal Industries Tetrafluor Division's Tetralon 720 materials have been used with acceptable results. Other PTFE materials could also be used, but qualification tests should be conducted before approval is made.

4.1.3 Electrical Insulation and Potting Compounds

Although specific tests have not yet been run, it is believed that standard electrical insulations and potting compounds capable of 300°F environmental temperature, other than silicone rubbers, are compatible with the CTFE fluid.

4.2 Pumps and Motors

In discussions with hydraulic pump and motor suppliers, concern was expressed that, due to the higher density and absolute viscosity of the AO-8 CTFE fluid, performance and life of their units operating with that fluid would compare unfavorably in the following respects with units of the same design operating with MIL-H-5606 fluid:

a. Efficiency

Increased flow losses into and out of a pump or motor, increased power loss, and lower efficiency were anticipated. However, data taken during tests of a modern aircraft pump did not bear this out. Delivery flow, case drain flow, power loss, and efficiencies were nearly identical to that measured with another pump of the same model operating in MIL-H-5606 fluid.

b. Inlet Pressure Requirements

Higher inlet pressure requirements were anticipated. This was born out in actual testing wherein, for example, the critical inlet pressure measured at the pump's rated inlet speed of 7,000 rpm and at 240°F inlet fluid temperature was 58.5 psia compared to 24.5 psia for the same model pump operating with MIL-H-5606 fluid under the same conditions.

c. Bearing Life

Potential bearing life problems necessitating larger envelopes, greater weight, lower speeds, and higher power losses were anticipated. During tests, this was realized. Lubrication failure occurred at the cylinder block to valve plate interface while operating at 7,000 rpm rated speed. Therefore, it appears that redesign or pump rebalancing may be required; or, as an option, that rated operating speed be reduced. Additional analysis and testing would be required to establish actual values.

d. Valve Plate Erosion

Valve plate erosion due to higher kinetic energy in fluid jets caused by the higher density of the CTFE fluid was anticipated. However, no valve plate erosion with CTFE fluid has been experienced in pump testing to date.

## 4.3 Hydraulic Servoactuators

The two fluid properties which will primarily affect servoactuator performance are density and bulk modulus.

Fluid density affects the valve flow and pressure drop per the relationship:

$$\frac{Q^2}{\Delta P} = \frac{K A^2}{\rho} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (26)$$

which comes from the classic equation: $Q = C_d A\sqrt{2\Delta P/\rho}$ . . . . . . . . . (27)

where:

$K$ = constant

$A$ = valve metering slot area

$C_d$ = valve slot orifice discharge coefficient

If valve gains (e.g. no-load flow rates) equivalent to that obtained with MIL-H-5606 fluid are desired with the higher density CTFE fluid, the valve slot areas may be increased for density variation through the formula:

$$\frac{A_2}{A_1} = \sqrt{\frac{\rho_2}{\rho_1}} \quad \text{therefore} \quad A_2 = \sqrt{\frac{1.836}{.84}} \cdot A_1 = 1.48A_1 \cdots \quad (28)$$

For the CTFE fluid, with its density of 1.836 compared to 0.84 for MIL-H-5606 fluid, the valve slot areas should be approximately 50 percent larger to obtain equivalent gain.

The fluid modulus has a primary affect upon the servoactuator's response to an oscillation at the valve input and actuator output. A servoactuator specification will generally require an exacting match between the command input and the output for oscillatory inputs up to those rates required for aircraft stability and/or maneuvers, and will require that the output to command-input ratio be greatly reduced when approaching the aerodynamic-flutter and the structure/actuator loop spring natural frequencies.

The bulk modulus of the CTFE fluid is approximately 12 percent lower than the MIL-H-5606 fluid values, and has the effect of slightly reducing the higher frequency output to command-input ratio and reducing the structure/actuator loop spring natural frequency. The former effect is a slight attribute but the reduced natural frequency effect decreases the margin between an acceptable actuator amplitude-ratio/frequency relationship and the structure/actuator loop spring natural frequency.

Since the worst-case analysis is the usual method for determining the dynamic condition acceptability of an actuator installation, the maximum fluid temperature is used. Bulk modulus is lowest at maximum temperature producing the softest fluid spring and the lowest surface-induced natural frequency. All lower fluid temperature have greater bulk moduli, therefore producing better response at higher frequencies. The bulk moduli of the CTFE fluid never exceeds that of MIL-H-5606 fluid at corresponding temperatures thus its amplitude ratio values will not exceed those of MIL-H-5606 fluid.

In a computer simulation study of a modern high-performance flight control servoactuation system, it was found that response very nearly equal to that obtained with MIL-H-5606 fluid could be obtained with the CTFE fluid if the valve slot areas were increased as previously noted. However, without such increase, substantial deviation was observed. See Reference 1.

## 4.4 Electrohydraulic Servovalves

The most significant property of the CTFE fluid in regard to servovalve performance, as compared to performance with MIL-H-5606 fluid, is its higher density.

For two-stage nozzle flapper valves, the flow rate will be reduced in relation to the square root of the density ratio of the two fluids, i.e.:

$$\frac{Q_2}{Q_1} = \sqrt{\frac{\rho_1}{\rho_2}} \quad \text{therefore} \quad Q_2 = \sqrt{\frac{.84}{1.836}}\; Q_1 = 0.67Q_1 \quad \cdots \quad (29)$$

This is correctable (within the confines of the valve) by increasing the metering slot width and/or the stroke of the second-stage spool. First-stage flow, which appears as internal leakage, will be reduced with a given set of orificing due to the fluid density increase. However, this reduction in open-center flow would reduce frequency response characteristics of the valve at its maximum opocrating amplitudes. Larger orifice sizes could be used to reestablish response characteristics, but the larger nozzle would lower the first-stage pressure and the resultant second-stage spool force gain.

Therefore it appears that valves of this type will require different internal sizing of metering slots, nozzles, orifices, and feedback springs in order to achieve rated flows and dynamic performance equivalent to that in MIL-H-5606 fluid systems. However, one valve supplier stated that this detail selection of internal sizing is the normal design process used to tailor valve performance to the requirements of a particular system regardless of the oil being used, and that, with one exception, the required internal sizing appears to be well within the limits of normal design. The exception is that rated flow might not be achievable in the desired valve envelope.

He also noted that both MIL-V-27162 and ARP-490 specify standard maximum envelopes and mounting flange patterns for the various size classes of servovalves, and that, there is a maximum practical flow that can be put through any one of these size classes. Because of the difference in densities, this maximum flow will be decreased by approximately one third for valves run on CTFE fluid. Thus, systems which require valves that are designed near their flow limit with MIL-H-5606 fluid, may require larger valves for use with CTFE fluid.

He also stated that although the higher absolute viscosity of the AO-8 CTFE fluid implies lower spool/sleeve leakage, this quantity is normally so small that the effect on valve total internal leakage will be negligible. The lower bulk modulus of CTFE fluid is not considered to be significant since entrained air generally limits the effective bulk modulus to a much lower number.

Jet pipe valves will probably require no modification to account for fluid density change, but may require some small increase in nozzle feedback diameter to reduce viscosity induced losses at extreme low temperatures with the higher viscosity CTFE fluids such as AO-8.

## 4.5 Flow and Pressure Control Valves

For other hydraulic system components such as flow and pressure control valves, fuses, etc., the CTFE fluid's density is the property primarily affecting design. These components contain orifices as their critical design feature, and orifice size is principally a function of the design flow rate, allowable pressure drop, and the fluid's density. The previous discussion regarding servoactuator valve slot width details those changes required when designing for performance comparative to that obtained with MIL-H-5606 fluid.

In valves subject to leakage from high to low pressure, some commonly used fluids (phosphate esters) have the potential to cause metal erosion of the metering lands or sealing edges. This erosion phenomenon, in turn, increases the leakage at such valves and the total system quiescent flow. If allowed to continue, the system response becomes sluggish and potentially dangerous from lack of controllability.

The valve erosion experienced with phosphate ester fluids is due to an electrochemical corrosion mechanism; and, it has been found that the electrical conductivity of those fluids causing the valve erosion damage falls within a particular range of values. In tests conducted on the evaluation program, it was found that the AO-8 CTFE fluid was sufficiently removed from the erosion-prevalent band to indicate that no electrochemical valve erosion potential exists. See Figure 11. In addition, the measured wall current is very low, even negative, which also indicates that the fluid should not be erosive. See Section 5.13 for details.

## 4.6 Hydraulic Fluid Reservoirs

Reservoirs are often the largest and heaviest single components in a hydraulic system; and, system designers may experience considerable difficulty in finding a satisfactory location for their installation. The following fluid properties should be considered by reservoir designers:

a. Thermal coefficient of expansion

Sufficient reservoir volume must be provided to accommodate the expansion of the total system fluid volume from the minimum design temperature to the maximum design temperature, e.g. from -65°F to 275°F for a Type II system. The coefficient of thermal expansion for CTFE fluid is 0.00050 compared to 0.00040 $in^3/in^3$/deg F for MIL-H-5606 fluid. For the complete temperature rise of 340 degrees F, a CTFE fluid reservoir must provide an expansion volume of 17 percent of the total system fluid volume which is 25 percent greater than the 13.6 percent expansion volume required for MIL-H-5606 fluid reservoirs.

b. Bulk modulus

Sufficient fluid volume must be provided in the reservoir to accommodate the fluid compressed in the system pressure lines and components. The fluid volume required in the reservoir to make up volume compressed in the pressure manifold (pressure lines, etc.) when the system is pressurized to rated pressure is a function of the fluid's bulk modulus. Since the bulk modulus is the inverse of the fluid compressibility, the differential volume per unit volume is expressed by the equation:

$$\frac{\Delta V}{V} = \frac{\Delta P}{\beta} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (30)$$

where: V = volume of fluid pressurized
$\Delta V$ = reduction in fluid volume
$\beta$ = fluid bulk modulus

For a 3000 psi system at room temperature with MIL-H-5606 fluid,

$$\frac{\Delta V}{V} = \frac{3000}{273,300} = .010977 \quad \text{(ie, 1.1\%)}$$

and for CTFE fluid, $\frac{\Delta V}{V} = \frac{3000}{240,700} = .012464$ (ie, 1.25%)

The 0.15 percent difference is inconsequential.

c. Vapor pressure

The reservoir pressurization pressure must be high enough to prevent fluid vaporization and cavitation in the pump. The critical inlet pressure of a hydraulic pump, which is the minimum inlet pressure below which cavitation commences, is directly related to fluid vapor pressure; and, a fluid with a high vapor pressure may require a higher reservoir pressure than a low vapor pressure fluid. Fluid vapor pressure increases with increased temperature. Therefore, although other temperature conditions may also require examination depending upon the suction line, pump, and flight temperature profile parameters, the maximum operating temperature condition should always be evaluated. The maximum pump inlet temperature for a 275°F fluid discharge temperature would be approximately 225°F.

As noted below, the vapor pressure for the AO-8 CTFE fluid at that temperature is nearly equal to and somewhat lower than for the MIL-H-5606 fluid.

| FLUID | VAPOR PRESSURE @ 225F |
|---|---|
| MIL-H-5606 | 14.25 mm of Hg = .276 psi |
| AO-8 CTFE | 10.5 mm of Hg = .203 psi |

This would indicate that no significant change in reservoir pressurization would be required. However, as noted in Section 4.2.b, the critical inlet pressure measured with a typical pump was considerably higher with the AO-8 CTFE fluid indicating that higher reservoir pressure is required. This is attributed to the higher inlet pressure losses due to the relatively higher density and viscosity of the AO-8 CTFE fluid.

d. Bubble collapse rate

A low collapse rate could indicate a foaming tendency which could dictate the use of a separated type reservoir design.

A detriment to many systems in the past is a fluid's tendency to form foam when entrained and dissolved air form free air bubbles. If large volumes of foam develop in the distribution (plumbing) system, the fluid volume displaced must be accommodated in the reservoir or dumped overboard. Foam entering a hydraulic pump inlet line can have disasterous effects upon lubrication and output flow. The design of the fluid inlets and outlets of a reservoir (especially a non-separated type) can have a great deal to do with the foam generation. Separated reservoirs don't entirely eliminate the foaming problem, however, as air may enter the system through system seals and during component replacement.

The AO-8 CTFE fluid was tested for foaming tendency and it passed the MIL-H-5606 requirements. However, during pump testing, foaming in the reservoir was observed when the reservoir was depressurized following pump shutdown. This was apparently due to air coming out of solution in the fluid upon relief of the relatively high (80 psig) reservoir pressure required to maintain the suction line above the high critical inlet pressure. Additional investigation to determine the comparative air solubility of the AO-8 CTFE fluid as well as the MIL-H-5606 and MIL-H-83282 fluids is advised.

## 4. Heat Exchangers

Nearly all modern aircraft hydraulic systems require a heat exchanger to stabilize the maximum fluid temperature. Excessively high temperatures can rapidly deteriorate the hydraulic fluid as well as the elastomeric seals. The fluid properties most concerned with heat exchanger sizing (heat transfer area) are specific heat (heat capacity) and thermal conductivity.

As shown in Section 5.5, the specific heat of the CTFE fluid is approximately half that of MIL-H-5606 fluid. Therefore, with a specific gravity approximately double that of MIL-H-5606 fluid, the heat capacity of a CTFE fluid system would be nearly equal to that of a MIL-H-5606 fluid system as shown below.

$$Q = C\,m\,\Delta T \qquad (31)$$

where:

$Q$ = quantity of heat, BTU
$C$ = coefficient of specific heat or heat capacity, BTU/lb/°F
$m$ = mass, $lb_m$
$\Delta T$ = temperature differential, °F

Thus, if it is assumed that the maximum temperatures of both systems are equal, then

$$\frac{Q_2}{Q_1} = \frac{C_2}{C_1}\,\frac{m_2}{m_1} \qquad (32)$$

Then, if Q and m are per unit volume, $\rho = \frac{m}{v}$ and $Q' = \frac{Q}{v}$

$$\frac{Q_2'}{Q_1'} = \frac{C_2}{C_1}\,\frac{\rho_2}{\rho_1} \qquad (33)$$

At maximum system bulk fluid temperature of 275°F, the ratio of heat capacities for a CTFE system compared to a MIL-H-5606 system is:

$$\frac{Q_2'}{Q_1'} = \frac{.255}{.55}\quad\frac{1.68}{.77} = 1.012$$

The other primary fluid property in heat exchanger design is the thermal conductivity. The thermal conductivity of the CTFE fluid and MIL-H-5606 fluid is shown in Section 5.6. Heat exchanger sizing generally follows the thermal conductivity coefficient when comparing fluids if their heat capacities per unit volume and fluid flow rates are similar. This may be shown from the equation:

$$\frac{Q}{\Delta t} = k \; A \; \Delta T \qquad (34)$$

where:

$Q$ = quantity of heat, BTU
$\Delta t$ = time period, hr.
$k$ = thermal conductivity coefficient, BTU/hr/ft$^2$/°F/ft
$A$ = heat exchanger plate area, ft$^2$
$\Delta T$ = temperature differential, °F

Therefore, when comparing fluids, the time element, $\Delta t$, remains the same, as does the temperature, $\Delta T$, since it is desired that the maximum system temperatures remain equal. Then,

$$\frac{Q_2}{Q_1} = \frac{k_2}{k_1}\frac{A_2}{A_1} \quad \text{or} \quad \frac{A_2}{A_1} = \frac{Q_2}{Q_1}\frac{k_1}{k_2} \qquad (35)$$

Since the heat capacity of the two fluids doesn't vary appreciably (as shown previously) and the system heat generation is not expected to be significantly different, then,

$$\frac{A_2}{A_1} = \frac{k_1}{k_2} \qquad (36)$$

The thermal conductivity of the CTFE fluid is approximately one-half that of MIL-H-5606 fluid; therefore, the size of a heat exchanger in a system utilizing CTFE fluid must be approximately twice the size required for a MIL-H-5606 fluid system.

## 4.8 Hydraulic Filters

Hydraulic system particle contamination is controlled by the use of mechanical screens or filters of the pleated element type using a treated paper, woven wire, or combination thereof as the filtering medium. For determination of the medium's (element) surface area, the Darcy relationship of laminar flow through a multiplicity of straight, constant-diameter, capillary passages is used.

$$Q = \frac{\kappa N \pi d^2 \Delta P}{A \mu T} \qquad (37)$$

where:

$\kappa$ = permeability constant
$Q$ = hydraulic fluid flow
$N$ = number of flow passages
$d$ = capillary diameter
$T$ = length of capillary passages
$\mu$ = absolute viscosity
$\Delta P$ = differential pressure
$A$ = filter media area

For an equivalent system using CTFE fluid as compared to MIL-H-5606, K, Q, $\Delta P$, T and d will remain equal and $Nd^2$ is proportional to element area.

$$\text{Then} \quad \frac{A_2}{A_1} = \frac{\mu_2}{\mu_1} \quad \text{(38)}$$

Thus, the CTFE fluid filters should be somewhat larger than the MIL-H-5606 filters in an equivalent system; and, the actual increase will depend upon the design temperature.

It is difficult to assess the filter life aspect but several fluid properties can have an effect upon the amount of contaminant generated by the fluid or system. The fluid-generated particulate matter is the result of fluid decomposing thermally, oxidatively, catalyticly, chemically and/or by mechanical shear. In many of the fluid degradation aspects, the CTFE fluid excells as it is basically a very stable inert material.

System-generated contamination is the result of normal system fluid replenishment, component mechanical wear, seal wear and component replacement. Of these, only fluid replenishment, component wear, and seal wear are related to the fluid properties. Contamination during fluid replenishment is controlled by the quality control required by the fluid specification, the filling techniques and equipment, and the filtration (if any) in the filling circuit. Most of the component-generated metal wear particles originate within the hydraulic pumps. Elastomeric and plastic pieces are "nibbled" or torn away from seals or anti-extrusion devices such as backup rings and cap rings.

No appreciable difference in contamination generation between CTFE fluid and MIL-H-5606 fluid was observed in testing to date. Since the CTFE fluid is expected to have a greater endurance life without fluid breakdown, no increase in required filter size from that aspect is anticipated. However, great care should be taken to avoid mixing the two fluids.

## 5.0 FLUID PROPERTIES

### 5.1 Composition

CTFE fluids are saturated low molecular weight polymers of chlorotrifluoroethylene having the general formula $(CF_2\ CF\ Cl)_n$. The Halocarbon Products' AO-8 CTFE fluid tested under the referenced contract is a colorless basestock oil with the following additives:

(a) viscosity index (V.I.) improver: a copolymer of chlorotrifluoroethylene and vinylindene fluoride, 5% by weight. This is a basic additive in the AO-8 fluid as received from Halocarbon Products.

(b) lubricity and anti-wear additive: Molyvan A (molybdenum oxysulphide dithiocarbamate) less than 1% by weight. This additive was added by AFWAL/MLBT and was used in fluid utilized in the long-term pump test and the servoactuator test in the referenced contract. For this report, that fluid is designated as AO-8MVA.

The inclusion, deletion, or replacement of these additives will be defined in future Air Force Programs.

The AO-8 CTFE fluid has a density of 1.836 gm/cm$^3$ at 77°F which is over twice that of the standard petroluem-based hydraulic fluid per specification MIL-H-5606. Its absolute viscosity is nearly equal that of the MIL-H-5606 fluid in the normal operating temperature range (100-250°F) but more than double that of MIL-H-5606 fluid at -65°F. As shown in the following sections, most of its other hydraulic fluid properties are comparable to MIL-H-5606 fluid. In addition, it offers the advantage of chemical inertness toward practically all compounds and solutions, and is nonflammable.

Comparative viscosity data on two lower-viscosity versions of the CTFE fluid was provided during the program by AFWAL/MLBT. Those fluids are Halocarbon AO-2 and Halocarbon 1.8/100, and neither contain a V.I. improver.

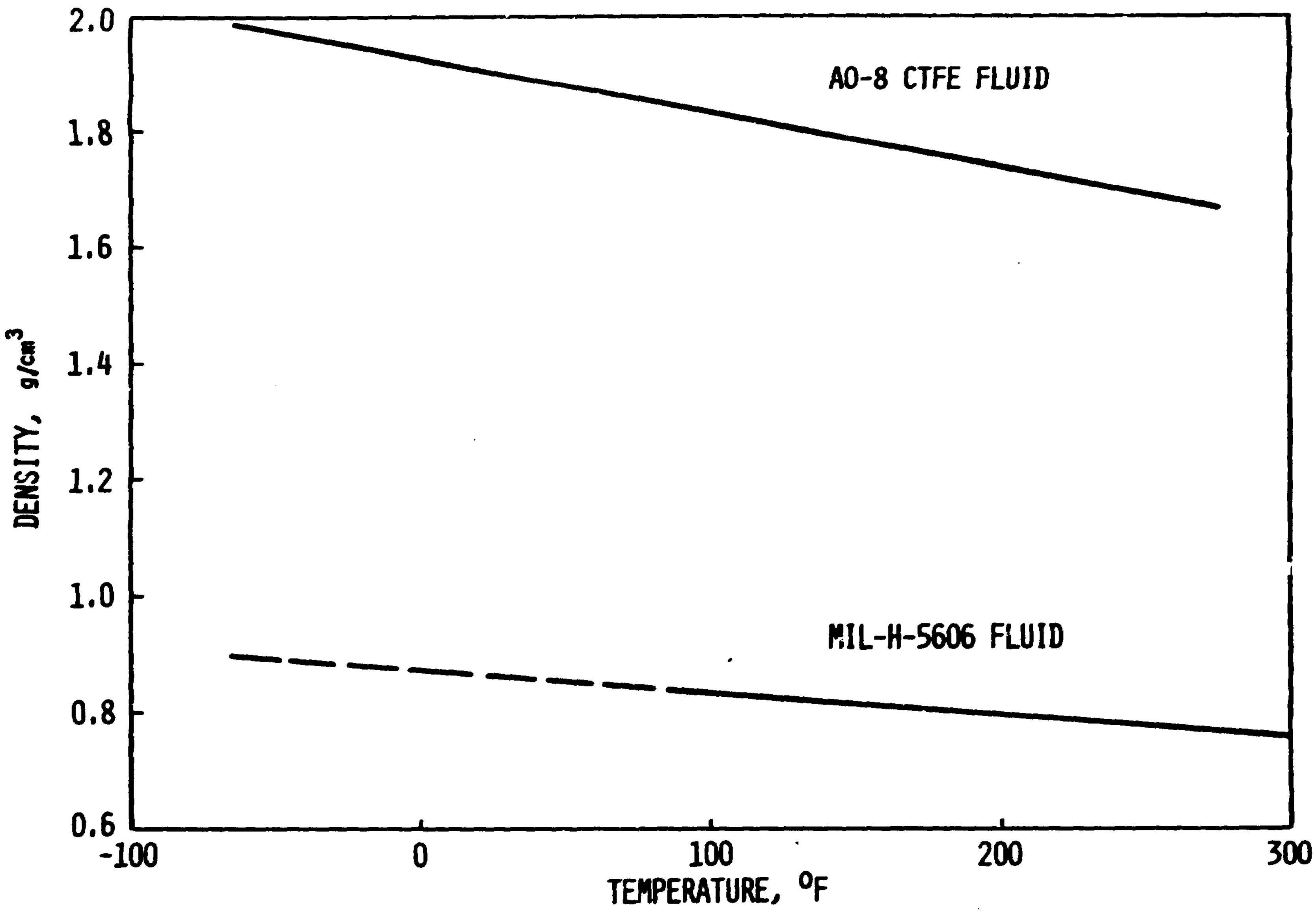


Figure 3. Density of AO-8 CTFE fluid and MIL-H-5606 fluid

## 5.2 Density and Coefficient of Thermal Expansion

The density versus temperature curves for the AO-8 CTFE fluid and MIL-H-5606 fluid are as shown in Figure 3. It is expected that the density of other CTFE fluids will be the same as the AO-8 fluid. The following values are given for reference:

| Property | CTFE | MIL-H-5606 | Source |
|---|---|---|---|
| Density @ 77°F (gm/cc) | 1.836 | 0.84 | AFML data |
| Coefficient of Thermal Expansion ($in^3/in^3$/degF) | 0.0005 | 0.0004 | Calculated values per ASTM D1903 (Ref. 5) |

## 5.3 Viscosity

Curves of kinematic viscosity, in centistokes at atmospheric pressure, versus temperature for the AO-8 CTFE fluid, for the lower-viscosity AO-2 and 1.8/100 CTFE fluids, and for MIL-H-5606 fluid are shown in Figure 4. The AO-8 curve is based upon AFWAL/MLBT data, the AO-2 and 1.8/100 curves were provided by the Halocarbon Products Corporation, and the MIL-H-5606 curve is from SAE AIR 1362 (Reference 3).

In addition, curves of absolute viscosity, in centipoises at atmospheric pressure, versus temperature for the four fluids are shown in Figure 5.. These curves were derived from the kinematic viscosity and density values per the following relationship:

$\mu = \rho\nu$ where $\mu$ = absolute viscosity (centipoise)

$\rho$ = mass density (gm/cc)

$\nu$ = kinematic viscosity (centistoke)

These curves show that, for fluid flow calculations which require the use of the absolute viscosity values, the AO-8 CTFE fluid has significantly greater low-temperature viscosity than the MIL-H-5606 fluid.

5. ANSI/ASTM D1903-63 (Reapproved 1978), Standard Test Method for Coefficient of Thermal Expansion of Electrical Insulating Liquids of Petroleum Origin, and Askarels, American Society for Testing and Materials, Philadelphia, PA

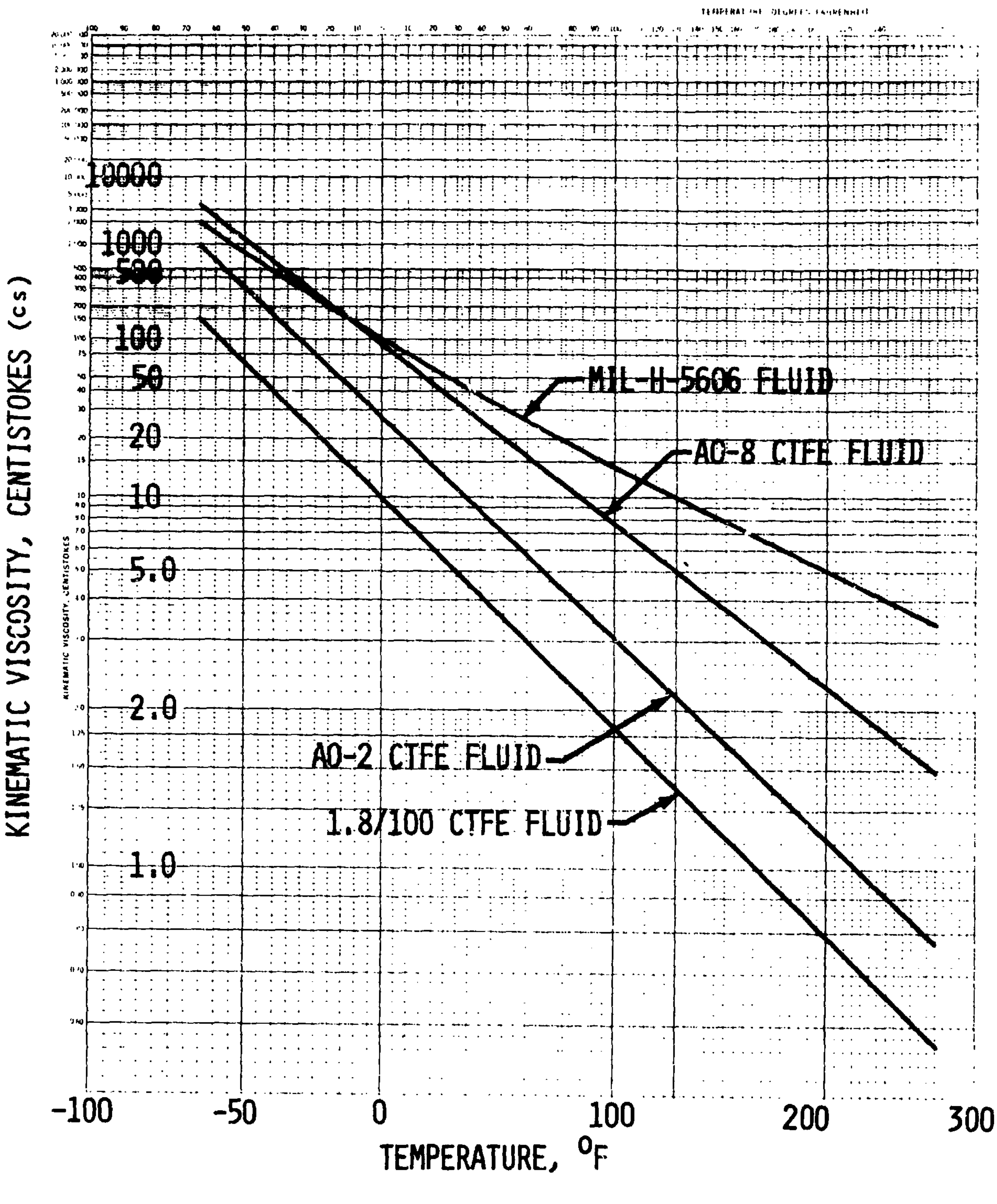


**Figure 4. Kinematic viscosity of three CTFE fluids and MIL-H-5606 fluid**

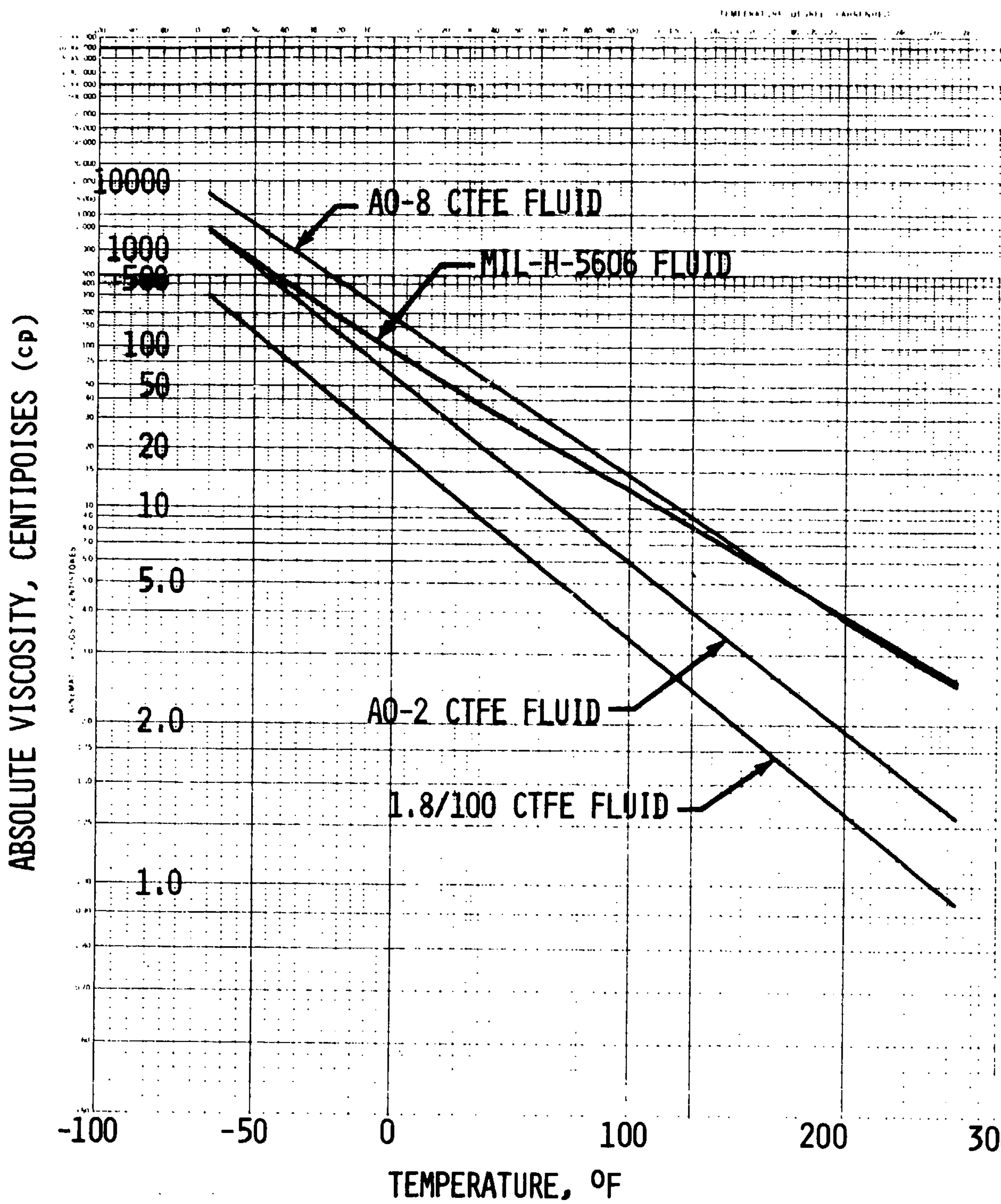


Figure 5. Absolute viscosity of three CTFE fluids and MIL-H-5606 fluid

The following viscosity values, both kinematic and absolute, are presented for reference. Note that, for MIL-H-5606 fluid, typical values per AIR 1362 and specified limits per MIL-H-5606C are listed.

| Temperature degrees F | Kinematic Viscosity in Centistokes (cs) | | | | |
|---|---|---|---|---|---|
| | MIL-H-5606 | | AO-8 CTFE | AO-2 CTFE | 1.8/100 CTFE |
| | (AIR 1362) | (LIMITS) | (AFWAL/MLBT) | (Halocarbon) | (Halocarbon) |
| -65 | 2000 | 3000 max | 3100 | 900 | 150 |
| -40 | 440 | 500 max | | 165 | 43 |
| 100 | 14 | 14 min | 7.6 | 3.1 | 1.9 |
| 210 | 5 | 5 min | 2.2 | 1.1 | 0.8 |
| 275 | 3.4 | | 1.5 | 0.77 | 0.57 |

| Temperature degrees F | Absolute Viscosity in Centipoises (cp) | | | | |
|---|---|---|---|---|---|
| | MIL-H-5606 | | AO-8 CTFE | AO-2 CTFE | 1.8/100 CTFE |
| -65 | 1800 | 2700 max | 6176 | 1790 | 300 |
| -40 | 400 | 450 max | | 325 | 85 |
| 100 | 12 | 12 min | 14 | 5.7 | 3.5 |
| 210 | 4 | 4 min | 3.8 | 1.9 | 1.4 |
| 275 | 2.6 | | 2.5 | 1.3 | 0.93 |

## 5.4 Bulk Modulus

The curves of adiabatic tangent bulk modulus for the AO-8 CTFE fluid and for MIL-H-5606 fluid are as shown in Figures 6 and 7 respectively. It is expected that the bulk modulus of other CTFE fluids will be same as the AO-8 fluid. The following values (in psi) are given for reference:

| | AO-8 CTFE FLUID (Boeing Data) | | | MIL-H-5606 FLUID (SAE AIR 1362) | | |
|---|---|---|---|---|---|---|
| PRESSURE | 72.5°F | 150°F | 250°F | 72°F | 150°F | 250°F |
| Zero psig | 206,207 | 150,063 | 99,225 | 250,000 | 195,000 | 133,000 |
| 1500 psig | 225,661 | 171,672 | 116,746 | 268,000 | 213,000 | 151,000 |
| 3000 psig | 245,029 | 191,442 | 137,224 | 286,000 | 231,500 | 170,000 |
| 4500 psig | 264,274 | 211,779 | 159,135 | 305,000 | 250,000 | 190,000 |

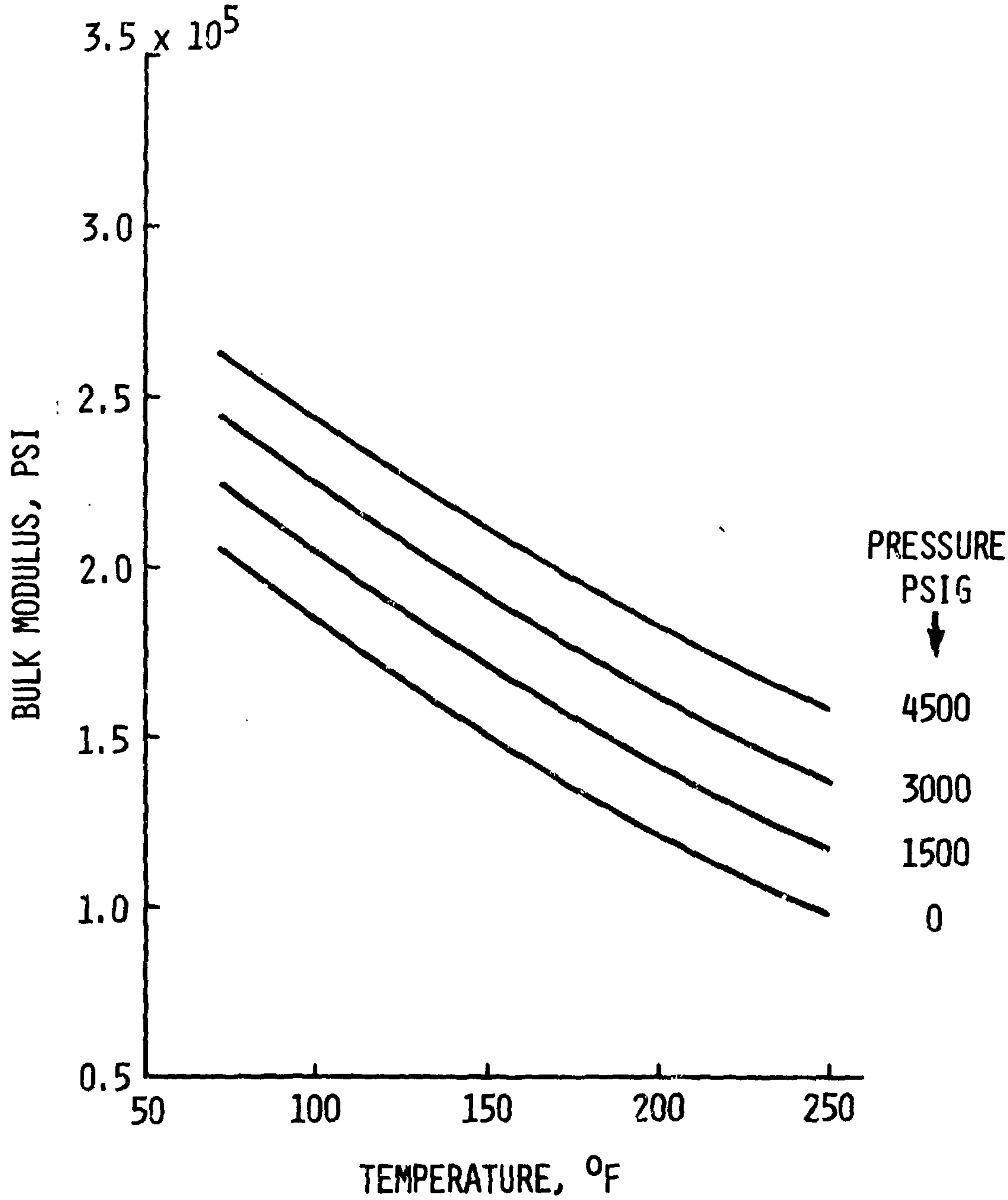


Figure 6. Adiabatic tangent bulk modulus of AO-8 CTFE fluid

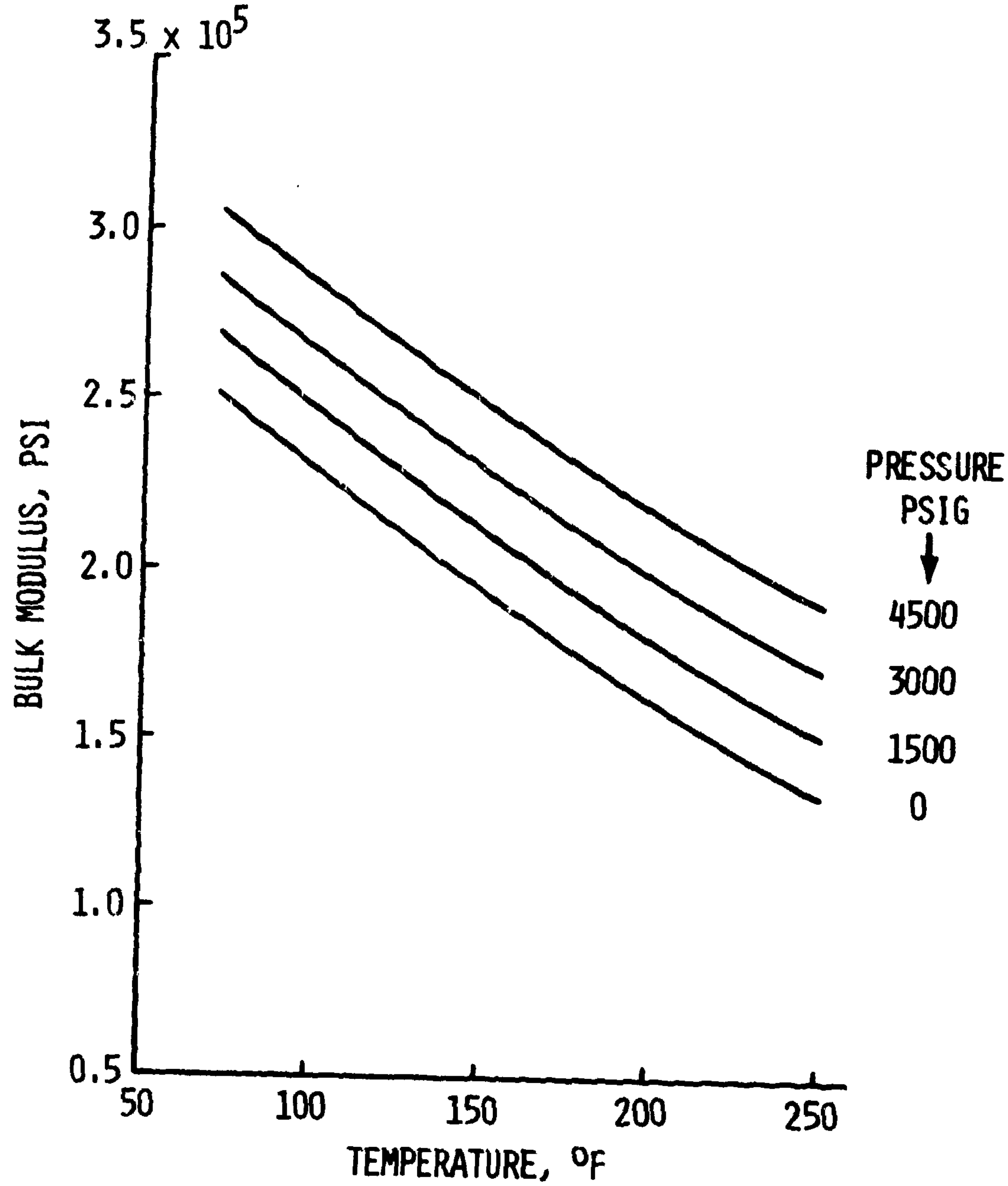


Figure 7. Adiabatic tangent bulk modulus of MIL-H-5606 fluid

## 5.5 Specific Heat (AFWAL/MLBT Data)

The curves of specific heat for the AO-8 CTFE fluid and for MIL-H-5606 fluid are as shown in Figure 8. It is expected that the specific heat of other CTFE fluids will be the same as the AO-8 fluid. The following values are given for reference:

| Temperature | Specific Heat (BTU/lb/°F) AO-8 CTFE | Specific Heat (BTU/lb/°F) MIL-H-5606 |
|---|---|---|
| 100°F | 0.234 | 0.4609 |
| 200°F | 0.246 | 0.5316 |

## 5.6 Thermal Conductivity (AFWAL/MLBT Data)

The curves of thermal conductivity for the AO-8 CTFE fluid and for MIL-H-5606 fluid are shown in Figure 9. It is expected that the thermal conductivity of other CTFE fluids will be the same as the AO-8 fluid. The following values are given for reference:

| Temperature | Thermal Conductivity BTU/hr/ft$^2$/°F/ft AO-8 CTFE | Thermal Conductivity BTU/hr/ft$^2$/°F/ft MIL-H-5606 |
|---|---|---|
| 100°F | 0.043 | 0.0649 |
| 200°F | 0.039 | 0.0579 |

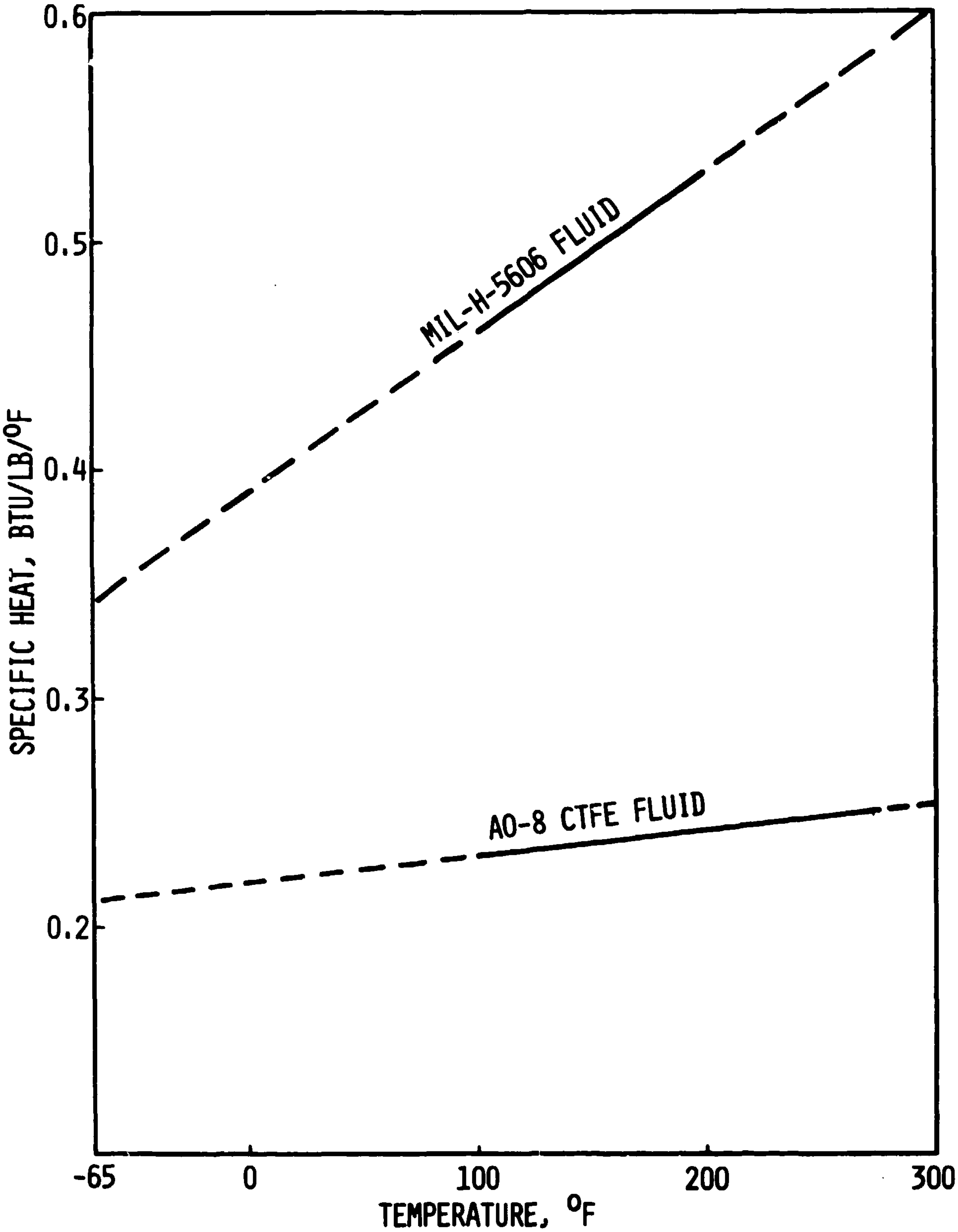


**Figure 8. Specific heat of AO-8 CTFE fluid and MIL-H-5606 fluid**

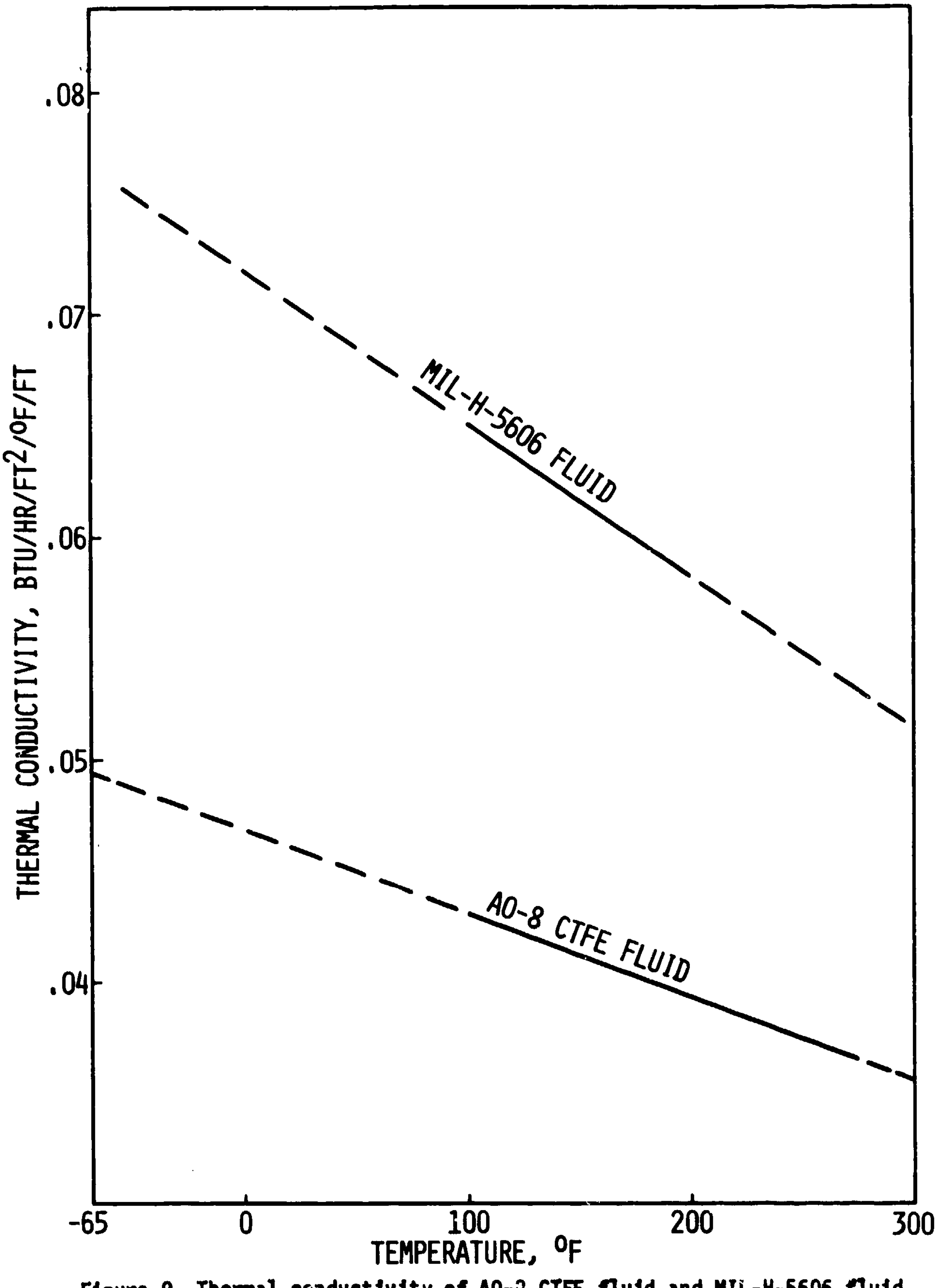


**Figure 9 Thermal conductivity of AO-3 CTFE fluid and MIL-H-5606 fluid**

## 5.7 Vapor Pressure

The curves of vapor pressure for the AO-8 CTFE fluid and for MIL-H-5606 fluid are as shown in Figure 10. Higher vapor pressures can be expected with lower-viscosity CTFE fluids. The following values are given for reference:

| | Vapor Pressure (mm Hg) | |
|---|---|---|
| | AO-8 CTFE (AFWAL/MLBT Data) | MIL-H-5606 (Mobil Oil Data) (Ref. 6) |
| Temperature | | |
| 210°F | 6 | 9.5 |
| 240°F | 15 | 19 |
| 300°F | 71 | 56 |

## 5.8 Foaming Tendency (AFWAL/MLBT Data)

The AO-8 CTFE fluid was tested per ASTM Method D892 (Reference 7) as specified in MIL-H-5606C. The foaming characteristic did not exceed the following limits as required to pass the test.

| Test | Foaming Tendency | Foam Stability |
|---|---|---|
| | Foam volume, ml, at end of 5-minute blowing period. | Foam volume, ml, at end of 10-minute settling period. |
| At 75°F | 65 ml (max) | Complete Collapse |

6. J. L. Herr and N. J. Pierce, Evaluation of MLO-68-5 Less Flammable Hydraulic Fluid, ASD-TR-70-36, McDonnell Aircraft Company, St. Louis, MO, September 1970.

7. ANSI/ASTM D892-74, Standard Test Method for Foaming Characteristics of Lubricating Oils, American Society for Testing and Materials, Philadelphia, PA.

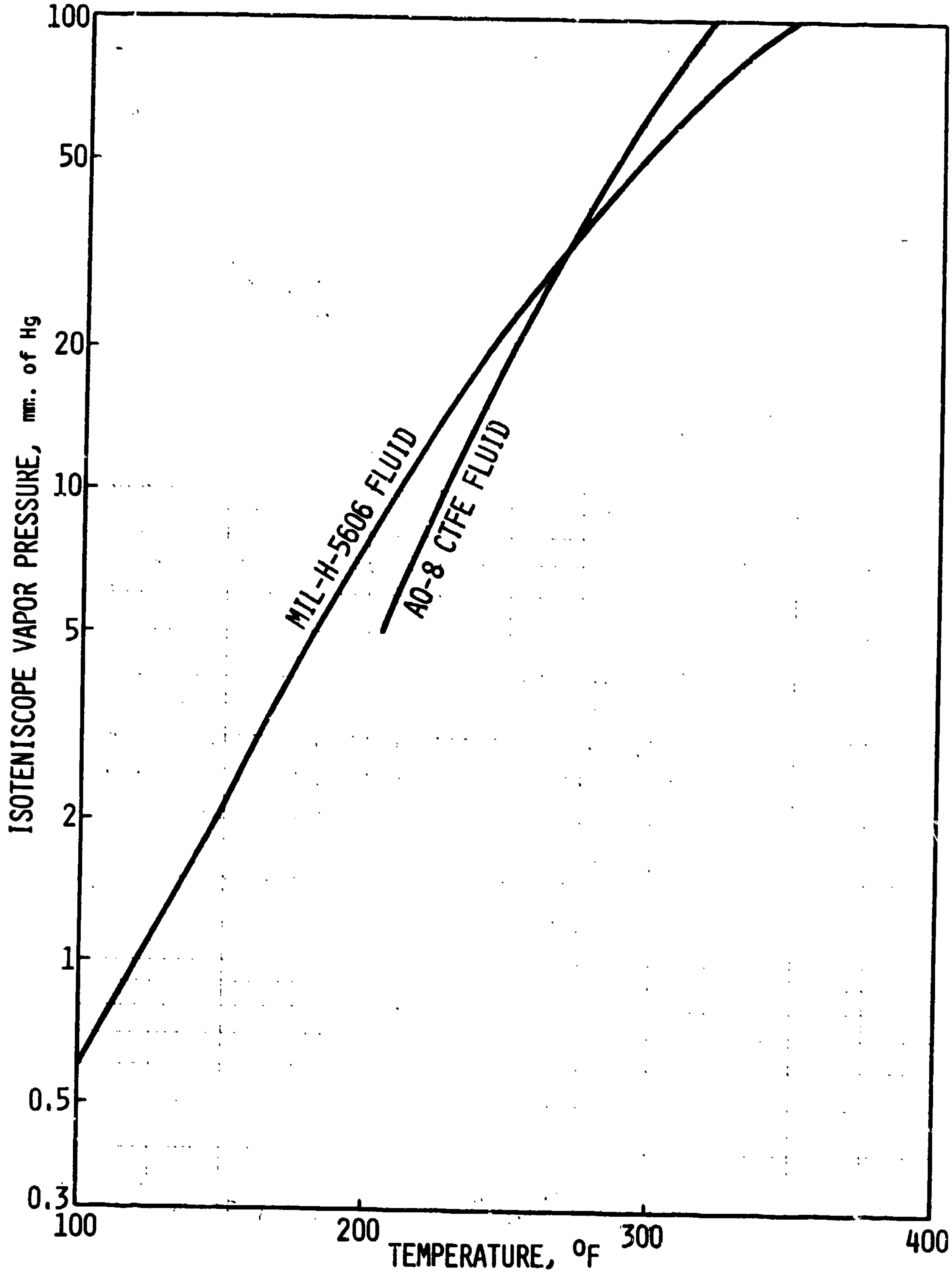


Figure 10. Vapor pressure of AO-8 CTFE fluid and MIL-H-5606 fluid

5.9 Flammability (Typical Values) (AFWAL/POSH Data. See Reference 8.)

| | Flammability Property | Target Value | AO-8 CTFE | MIL-H-5606 |
|---|---|---|---|---|
| 5.9.1 | Flash Point | - | None | 200 - 225°F |
| 5.9.2 | Fire Point | - | None | 255°F |
| 5.9.3 | Autogenous Ingition Temperature | >1,300°F | 1,170°F (Transient small blue flame) | 450°F |
| 5.9.4 | Hot Manifold Ignition Temperature | >1,700°F | | |
| | (Spray Delivery) | | >1,700°F | 1,400°F |
| | (Burette Delivery) | | >1,700°F | 750°F |
| 5.9.5 | Heat of Combustion (BTU/lb) | <5,000 | 2,390 | 18,000 |

5.9.6 Atomized Fluid Flammability

The AO-8 CTFE fluid and the MIL-H-5606 fluid were tested to determine the ignitability and flame propagation characteristics of their aerosolized sprays generated with an oil burner type nozzle. When exposed to flame from a propane torch, the MIL-H-5606 fluid ignited and sustained combustion after removal of the torch. The AO-8 CTFE fluid did not ignite when the propane flame was passed through its spray.

8. Leo Parts, Assessment of the Flammability of Aircraft Hydraulic Fluids, AFAPL-TR-79-2055, Monsanto Research Corporation, Dayton Laboratory, Dayton, Ohio, July 1979.

## 5.10 Stability (AFWAL/MLBT Data)

### 5.10.1 Low Temperature Stability

No clouding or solids at any temperature down to -65°F.

AO-8 CTFE Fluid: Pass MIL-H-5606 Fluid: Pass

### 5.10.2 Thermal Stability

Change in properties after 72 hours at 325°F with a nitrogen atmosphere and with eleven typical component metals immersed (See Section 6.2 and Figure 12):

| Fluid | Change in Neutralization Nr. | Change in Viscosity at 100°F |
|---|---|---|
| | Requirement $\leq 0.2$ | Requirement $\leq \pm 5\%$ |
| AO-8 CTFE | Pass | Pass |
| AO-8MVA CTFE | Pass | Pass |
| MIL-H-83282 @ 450°F | Fail (.64) | Pass |

### 5.10.3 Oxidative Stability

Change in properties after 72 hours at 325°F with an air atmosphere and with eleven typical component metals immersed. (See Section 6.2 and Figure 12):

| Fluid | Change in Neutralization Nr. | Change in Viscosity at 100°F |
|---|---|---|
| | Requirement $\leq 0.2$ | Requirement $\leq \pm 5\%$ |
| AO-8 CTFE | Pass | Pass |
| AO-8MVA CTFE | Pass | Pass |
| MIL-H-5606 | Pass | Pass |

5.10.4 Hydrolytic Stability

Change in properties after 72 hours at 325°F, with a nitrogen atmosphere, containing 0.2% water, and with eleven typical component metals immersed (See Section 6.2 and Figure 12):

| Fluid | Change in Neutralization Nr. | Change in Viscosity at 100°F |
|---|---|---|
| | Requirement $\leq$ 0.2 | Requirement $\leq \pm 5$ |
| AO-8 CTFE | Pass | Pass |
| AO-8MVA CTFE | Pass | Pass |
| MIL-H-5606 | Pass | Pass |

5.10.5 Shear Stability

Change in properties after testing in a magnetostrictive sonic oscillator per ASTM D2603 (Reference 9) were as follows. The data indicates that the AO-8 CTFE fluid is considerably more shear stable than MIL-H-5606 fluid.

| Fluid | Change in Viscosity at 100°F | Change in Viscosity at -40°F |
|---|---|---|
| AO-8 CTFE | +0.38% | -1.4% |
| MIL-H-5606 | -12.4% | -10.5% |

9. ANSI/ATSM D2603-76, Standard Test Method for Sonic Shear Stability of Polymer-Containing Oils, American Society for Testing and Materials, Philadelphia, PA.

## 5.11 Lubricity (AFWAL/MLBT Data)

Results of Shell Four-Ball Tests at 1200 rpm for one hour at 167°F.

| Fluid | Wear Scar 10 kg load | Wear Scar 40 kg load |
|---|---|---|
| AO-8 CTFE | 0.37 mm | 1.2 - 2.2 mm |
| AO-8MVA CTFE | 0.37 mm | 0.55 mm |
| MIL-H-5606 | 0.50 mm | 1.0 mm |

## 5.12 Valve Stiction Tendency (Boeing Data)

The CTFE fluid appears to deteriorate without forming substances such as varnish which could seize a spool valve or other small-clearance sliding surfaces. In an 800-hour test wherein a close-fitting spool valve was immersed in fluid which was cycled thermally through a temperature range of 100°F to 300°F, the maximum breakout friction was below the target maximum value of five pounds. Comparative values are as follows:

| Fluid | Maximum Slide Force |
|---|---|
| AO-8 CTFE | 1.0 lb |
| MIL-H-5606 | less than 0.1 lb |

## 5.13 Electrical Conductivity and Other Properties Related to Electrochemical Corrosion Wear of Hydraulic Valves

The following fluid properties are related to the cause for unacceptable wear that has occurred in hydraulic spool valves on commercial jet aircraft using phosphate ester base hydraulic fluids. Values are given for the phosphate ester base stock fluid, which caused such wear, and for the new Type IV phosphate ester fluid, which has reduced such wear, as well as for the CTFE fluid and MIL-H-5606 fluid. Reference 10 provides an explanation of the electrochemical wear mechanism and a description of a series of experiments which were performed to confirm that other possible wear mechanisms were not operative.

5.13.1 Electrical Conductivity (Electrochemical Technology Data)

| Units | Phosphate Ester Fluids Base Stock | Phosphate Ester Fluids Type IV | AO-8 CTFE Fluid | MIL-H-5606 Fluid |
|---|---|---|---|---|
| (mho/cm) | $2 \times 10^{-8}$ | $1 \times 10^{-6}$ | $<8 \times 10^{-12}$ | $<1 \times 10^{-11}$ |

The prevalent band of values associated with valve electrochemical wear is shown in Figure 11. Fluids above the $2 \times 10^{-8}$ to $2 \times 10^{-7}$ mho/cm band, such as the new Type IV phosphate ester fluid, and those below that band, such as the CTFE fluid and the MIL-H-5606 fluid, are not expected to cause unacceptable wear rates.

5.13.2 Wall Current (Electrochemical Technology Data)

| Units | Phosphate Ester Fluids Base Stock | Phosphate Ester Fluids Type IV | AO-8 CTFE Fluid | MIL-H-5606 Fluid |
|---|---|---|---|---|
| (μA) at a flow rate of 10 $cm^3/s$ | 0.25 to 0.7 | -0.1 to +0.2 | -0.004 | +.0001 |

The fluids with the small and negative wall currents should not be erosive.

5.13.3 Threshold Corrosion Current Density (Electrochemical Technology Data)

| Units | Phosphate Ester Fluids Base Stock | Phosphate Ester Fluids Type IV | AO-8 CTFE Fluid | MIL-H-5606 Fluid |
|---|---|---|---|---|
| ($mA/cm^2$) | 1.0 | 2 to 4 | | |
| ($\mu A/cm^2$) | | | <0.06 | <0.4 |

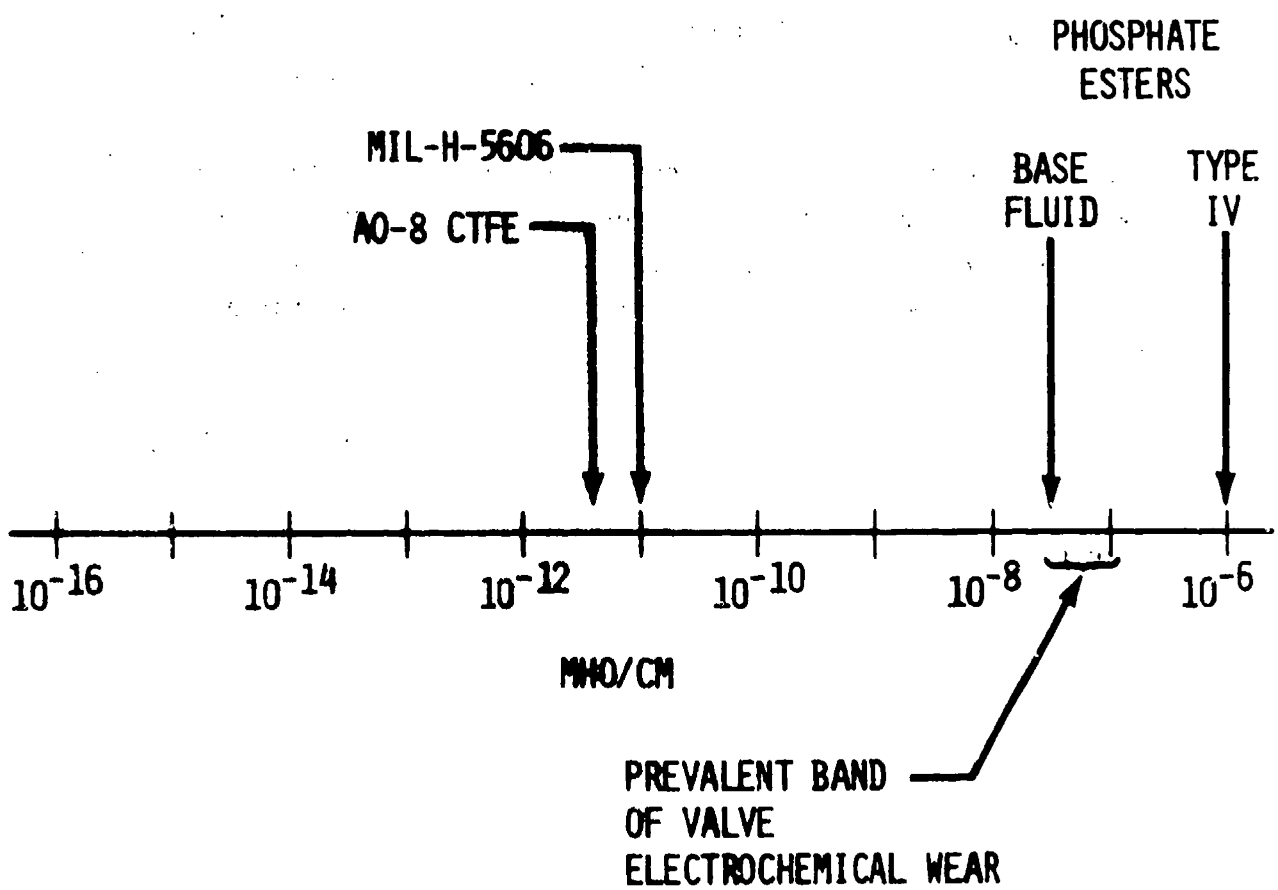


Figure 11. Electrical conductivity of various hydraulic fluids

The threshold corrosion current density of the CTFE fluid is smaller than for phosphate esters; but, with the small wall current, this should lead to no problem. Electrochemical wear of valves has not been a problem with MIL-H-5606 fluid.

### 5.13.4 Dielectric Breakdown

As a side observation, the CTFE fluid has a dielectric breakdown at a much lower electric field than phosphate esters. Phosphate esters withstand more than 400V (maximum voltage used) in the 0.005 cm gap in the NTP (needle-to-plane) cell or a field of 80,000 V/cm. The CTFE fluid had "avalanche" breakdown between 25V and 50V, or a field of 5,000 to 10,000 V/cm.

Threshold corrosion current density measurements of the MIL-H-5606 fluid were made with up to 400 volts across the cell without "avalanche" increase in conduction occurring. The cell current did slowly increase with time over a two-hour period, however, probably due to the formation of electrolysis products.

The significance of the "avalanche" breakdown of the CTFE fluid is yet to be evaluated.

10. T. R. Beck, D. W. Mahaffey, and J. H. Olsen, "Wear of Small Orifices by Streaming Current Driven Corrosion," Transactions of the ASME, December 1970, pp. 782-791.

## 6.0 MATERIAL COMPATIBILITY DATA

In the evaluation of a new fluid for aircraft application, it is absolutely necessary to determine each of the following:

a. its compatibility with other fluids in general aircraft use by determining the makeup and consistency of various fluid mixtures,

b. its compatibility with system and airframe materials in general use by determining if there is any chemical attack, corrosion, or other adverse effect under the full range of operating environments,

c. its compatibility with elastomeric seal materials to determine those commonly used materials which will be adversely weakened or otherwise harmed by exposure to the fluid under the full range of operating environments.

Such evaluations were conducted to some extent in the research program documented in Reference 1, and a number of compatibile and incompatible materials were identified. However, it must be recognized that laboratory testing has its limitations and that a continuous ongoing evaluation program must be conducted if the new fluid is to be successfully implemented.

### 6.1 Compatibility With Typical Component Materials (AFWAL/MLBT data)

The CTFE fluid appears to be compatible with most metals normally used in hydraulic components.

To determine the high-temperature stability of the fluid in a number of simulated system environments, and metal corrosion due to exposure to the fluid, a series of thermal, oxidative, and hydrolytic stability-corrosion tests were run in a rocking hydrogenation bomb in the presence of the following specimens of typical hydraulic component metals.

a. 52100 bearing steel ball
b. 4640 bronze disc
c. 3Al-2.5V titanium tube

d. 4340M steel disc

e. M-50 tool steel ball

f. 21Cr-6Ni-9Mn stainless steel tube

g. 440C stainless steel ball

h. 6061-T6 aluminum wafer

i. 15-5PH stainless steel disc

j. K6E cast iron ring

k. Nitralloy 135-M steel disc

In each of the stability tests, the bombs containing the test fluid and the metal specimens were heated to 325°F and rocked about a pivot for 72 hours with the fluid sloshing over the metal specimens. In the thermal stability-corrosion test, an atmospheric blanket of 100 psig dry nitrogen was maintained in the bomb; and, in the oxidative stability-corrosion test, 100 psig air was maintained. In the hydrolytic stability-corrosion test, a 0.2 percent volume concentration of water was mixed with the test fluid and 100 psig dry nitrogen maintained.

Following the tests, fluid properties were measured to determine the degree of breakdown, and, the metal specimens were weighed to determine the degree of corrosive attack or oxidation. The AO-8 CTFE fluid suffered no significant breakdown; and, as shown in Figure 12, the weight change of all metal specimens, except the 4640 bronze, the 52100 bearing steel, and the M-50 tool steel, was within allowable limits of $\pm 0.20$ mg/cm$^3$.

In addition, in a copper-strip corrosion test, the AO-8 CTFE fluid failed to meet the maximum tarnish/corrosion requirements specified for MIL-H-5606 fluid.

These limits which were exceeded cannot be taken as absolute no-go criteria, but the tests do indicate a greater reaction between the AO-8 CTFE fluid and the copper bearing alloys than was found with MIL-H-5606 fluid. This was also found in the hydraulic pump tests where a dark brown discoloration of the bronze parts was a common observation. So far, this reaction cannot be related to any part failures, but that is not to say that they won't occur after long term exposure.

| PARAMETER | TARGET VALUE | THERMAL-STABILITY TEST DATA | | | OXIDATIVE-STABILITY TEST DATA | | | HYDROLYTIC-STABILITY TEST DATA | | |
|---|---|---|---|---|---|---|---|---|---|---|
| TEST CONDITIONS TEMPERATURE | | 72-Hr With Dry Nitrogen 325F | 325F | 450F | 72-Hr With Dry Air 325F | 325F | 325F | 72-Hr, $N_2$ With 0.2% $H_2O$ 325F | 325F | 325F |
| FLUID PROPERTY CHANGE | | AO-8 CTFE | AO-8MVA CTFE | MIL-H 83282 | AO-8 CTFE | AO-8MVA CTFE | MIL-H -5606 | AO-8 CTFE | AO-8MVA CTFE | MIL-H -5606 |
| Neutralization No. | $\leq 0.2$ | <0.1 | <0.01 | 0.64 | <0.1 | <0.1 | 0.2 | <0.1 | <0.1 | <0.1 |
| Viscosity @ 100F | $\leq \pm 5\%$ | -2.4 | +2.0 | -1.0 | -1.9 | -2.7 | -1.0 | -2.3 | +2.8 | +1.1 |
| METAL WT. CHANGE | $\leq \pm 0.20$ $mg/cm^3$ | | | | | | | | | |
| 52100 Steel Ball | " | 0.00 | 0 | -0.34 | 0.00 | 0 | +0.13 | -0.22 | +0.24 | -0.56 |
| 4640 Bronze Disc | " | -0.22 | -0.40 | +0.60 | +0.30 | +0.02 | -0.08 | +0.54 | +0.36 | +0.05 |
| 3Al-2.5V Ti Tube | " | -0.01 | -0.03 | -0.02 | -0.03 | -0.03 | 0.00 | -0.03 | 0 | -0.01 |
| 4340M Steel Disc | " | +0.06 | -0.05 | -0.04 | -0.01 | -0.02 | +0.07 | -0.09 | +0.15 | -0.08 |
| M-50 Steel Ball | " | +0.01 | -0.04 | -0.44 | +0.01 | -0.02 | +0.13 | -0.25 | +0.22 | -0.45 |
| 21-6-9 Steel Tube | " | +0.02 | 0 | 0.00 | +0.01 | 0 | +0.04 | -0.05 | 0 | -0.02 |
| 440C Steel Ball | " | +0.01 | -0.02 | 0.00 | -0.04 | -0.06 | +0.05 | -0.04 | +0.16 | -0.01 |
| 6061-T6 Al Wafer | " | +0.01 | 0 | -0.01 | -0.02 | -0.02 | -0.02 | -0.01 | +0.06 | -0.04 |
| 15-5PH Steel Disc | " | +0.01 | -0.03 | +0.02 | +0.01 | -0.02 | +0.02 | -0.28 | +0.02 | -0.01 |
| K6E Cast Iron Ring | " | 0.00 | 0 | -0.18 | 0 | 0 | -0.01 | -0.07 | +0.08 | +0.03 |
| Nitralloy Steel Disc | " | +0.02 | -0.02 | -0.02 | 0 | 0 | +0.05 | -0.09 | +0.17 | -0.14 |

Figure 12. Fluid stability test data

It appears that, where alternate materials can be used, the usage of copper bearing alloys should be minimized. However, that restriction would be a serious penalty for many hydraulic components. The success of hydraulic pumps and motors, and actuator rod-gland bushings depends on having a good bearing alloy in the critical areas; and bronze is generally the best material. Therefore, its use should be evaluated thoroughly by testing under realistic load conditions.

The same can be said for carbon and graphite materials. Limited testing in a hydraulic pump indicated a potential problem.

6.2 Compatibility With Elastomers

As noted in Section 4.1.2, the Firestone phosphonitrilic fluoroelastomer (PNF) compound 280-001R, with Shore hardness of 80 durometer minimum, is the best elastomemr found to date for O-rings and other elastomeric seals intended for use in CTFE fluid. That conclusion is based upon extensive testing of candidate materials conducted by the Materials Laboratory and upon testing of actual seals conducted by the Boeing Company in the research program documented in Reference 1.

In the AFML tests, twelve seal materials were evaluated and the change in properties measured following aging for 72 hours at 275°F. On the basis of those tests, the PNF material was recommended as the most likely candidate. However, it was found that after aging it suffered a 50% loss of tensile strength (as did several of the other materials) and its volumetric swell was rather high (22 to 40%).

Nevertheless, in total, it looked better than the other materials; and, standard sized O-rings were tested in three separate test programs. In the first, which was a test of typical linear actuating cylinder static and rod seals the PNF O-rings performed as well as Standard MS28775 Buna N nitrile seals in comparative tests with MIL-H-5606 fluid when compared on the basis of acceptable leakage levels. However, the PNF O-rings did appear to soften more; and, several were found to adhere to the bottom of the seal grooves.

During a long-term test of a 3000-psi variable-displacement axial-piston pump, run with AO-8MVA CTFE fluid under various qualification test conditions per MIL-P-19692C, over 700 hours of operation were accumulated. By the end of the test, all PNF seals had softened and the shaft seal leaked a black tarry substance to the outside of the pump.

During a cycling test of a hydraulic flight control servoactuator, run with AO-8MVA CTFE fluid under qualification test conditions, over 280 hours of operation and approximately 2,500,000 stroke cycles were accumulated. All PNF seals were slightly softer than when originally installed, but not as soft as those in the pump tests, and, some had suffered nibbling damage. The limited operation of the servoactuator at high temperature (2 hours at 250F) implies that the degree of softening of the PNF material depends upon the time it is exposed to high temperature.

A more serious problem was attributed to escessive swelling of the PNF material. Ten leakage failures, which were apparently due to excessive swell of diametral static seal O-rings under each of two small filter caps on the servoactuator, occurred. Five of those were with PNF seals: one due to O-ring extrusion and four due to buildup of sufficient force to crack the aluminum plug at the root of its male screw thread. Two similar failures occurred with Viton O-rings and three with CPE (chlorinated polyethylene) O-rings. Therefore, it was concluded that some seal cavities such as these will have to be redesigned to allow space for volumetric expansion of the seal material.

## 7.0 CLEANING PROCEDURE

The following cleaning procedure is recommended for hydraulic components, tubing, and test stands being prepared for use with CTFE hydraulic fluid.

1. Clean with Stoddard Solvent per Federal Specification P-D-680.
2. Blow dry and evacuation dry.
3. Rinse with CTFE fluid.

## REFERENCES


1. E. T. Raymond, D. W. Huling, and R. L. Shick, Fire Resistant Aircraft Hydraulic System, AFWAL-TR-80-2112, Boeing Military Airplane Co., Seattle, WA, (in press).

2. S. M. Warren and J. R. Kilner, Fireproof Brake Hydraulic System, AFWAL-TR-81-2080, Boeing Military Airplane Co., Seattle, WA, September 1981.

3. SAE AIR 1362, Aerospace Information Report, Physicial Properties of Hydraulic Fluids, Society of Automotive Engineers, Inc., Warrendale, PA, May 1975.

4. SAE ARP 24B, Aerospace Recommended Practice, Determination of Hydraulic Pressure Drop, Society of Automotive Engineers, Inc., Warrendale, PA, 1-31-68.

5. ANSI/ASTM D1903-63 (Reapproved 1978), Standard Test Method for Coefficient of Thermal Expansion of Electrical Insulating Liquids of Petroleum Origin, and Askarels, American Society for Testing and Materials, Philadelphia, PA.

6. J. L. Herr and N. J. Pierce, Evaluation of MLO-68-5 Less Flammable Hydraulic Fluid, ASD-TR-70-35, McDonnell Aircraft Company, St. Louis, MO, September 1970.

7. ANSI/ASTM D892-74, Standard Test Method for Foaming Characteristics of Lubricating Oils, American Society for Testing and Materials, Philadelphia, PA.

8. Leo Parts, Assessment of the Flammability of Aircraft Hydraulic Fluids, AFAPL-TR-79-2055, Monsanto Research Corporation, Dayton Laboratory, Dayton, Ohio, July 1979.

9. ANSI/ATSM D2603-76, Standard Test Method for Sonic Shear Stability of Polymer-Containing Oils, American Society for Testing and Materials, Philadelphia, PA.

10. T. R. Beck, D. W. Mahaffey, and J. H. Olsen, "Wear of Small Orifices by Streaming Current Driven Corrosion," Transactions of the ASME, December 1970, pp. 782-791.

## APPENDIX A

## THE ESTIMATED WEIGHT PENALTY FOR USING AO-8 CTFE FLUID IN A LARGE TWIN-ENGINE MILITARY TRANSPORT AIRCRAFT HYDRAULIC SYSTEM

As noted in Section 3.4 in this design guide, the use of the CTFE fluid will increase system weight due to its higher density, and due to the increased tubing sizes necessary to maintain pressure losses within desired limits. To determine the magnitude of weight increase for a typical aircraft system, the study summarized herein was made. It included estimates of the weight increases due to the increased line sizes necessary to accomodate the Halocarbon AO-8 CTFE fluid throughout the hydraulic systems of the Air Force/Boeing YC-14 Advanced Medium STOL Transport (AMST) aircraft. It also included the increased weight of fluid in the actuators, reservoirs, and other components.

The increase weight increments for those items are as follows:

| Items | MIL-H-5606 System Weight | AO-8 CTFE System Weight | Weight Increase |
|---|---|---|---|
| Pressure Lines (Wet) | 932 lb | 1,591 lb | 659 lb |
| Return Lines (Wet) | 606 | 1,496 | 890 |
| Fluid in Components | 373 | 811 | 438 |
| Total of these items | 1,911 lb | 3,898 lb | 1,987 lb |

The weight of the YC-14 MIL-H-5606 hydraulic power supply system, including distribution tubing to the actuation systems, is approximately 3,500 lb. The 1,987-lb weight increase to incorporate the AO-8 CTFE fluid represents a 57% increase in the weight of the hydraulic power supply system.

The total weight of the YC-14 hydraulic power and actuation systems, including the power supply system, distribution tubing, and all acutation systems, is approximately 7,200 lb. The 1,987-lb weight increase to incorporate the AO-8 CTFE fluid represents a 28% increase in the weight of the overall hydraulic power and actuation systems.

APPENDIX B

## THE REDUCTION IN HYDRAULIC TUBE SIZES AND WEIGHT OBTAINABLE WITH A LOWER VISCOSITY CTFE FLUID

As noted in Section 3.4 in this design guide, one method for reducing the weight penalty which would be incurred by designing a hydraulic system for use with the AO-8 CTFE fluid is to use a lower viscosity blend of the fluid in order to reduce tubing sizes and fluid volume. As noted in Section 5.1, the AO-8 fluid has a viscosity index improver additive to bring its viscosity up to the normal range for aircraft hydraulic fluids (similar to MIL-H-5606 fluid). The basestock fluid for AO-8 is the Halocarbon Products' 1.8/100 fluid which has a viscosity approximately one-sixth that of the AO-8 fluid at design temperatures around 50F. (See Figure 5.)

Since the thickness of hydrodynamic lubrication films and their load carrying ability are a direct function of the lubricating fluid's viscosity, units such as hydraulic pumps and motors which are lubricated by the hydraulic fluid must be designed with bearing areas large enough to sustain all operating loads with the fluid viscosity existent over the full range of operating temperatures. If hydraulic pumps and motors can be designed to operate satisfactorily with the 1.8/100 fluid, it could be considered a viable choice; and, system weight could be reduced since its lower viscosity would allow smaller size tubing than that required for CTFE fluids, such as AO-8, with viscosities comparable to hydraulic fluids currently in use.

### Tube Sizing Equations

The following equations are taken from Eq. (14) and Eq. (16) shown in Section 3.3.1 for flow losses for laminar flow and turbulent flow respectively.

For laminar flow: $$D = \left(\frac{s\nu QL}{3663\Delta P}\right)^{1/4} \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(B1)}$$

For turbulent flow: $D = \left(\frac{s\nu^{0.25}Q^{1.75}L}{1756\Delta P}\right)^{1/4.75}$ . . . . . . . . . . . . . (B2)

where:

- $D$ = Tube inside diameter (in)
- $s$ = Fluid specific gravity, dimensionless, or Fluid density, (g/cm$^3$)
- $\nu$ = Fluid kinematic viscosity (cs)
- $Q$ = Fluid flow rate (gpm)
- $L$ = Length of tube (ft)
- $\Delta P$ = Pressure loss (psi)

To determine which equation to use, it is necessary to first calculate the Reynolds number. Any one of the four equations noted in Section 3.3.1, Eq. (5), Eq. (6), Eq. (7), or Eq. (8), may be used depending upon the fluid flow parameters which are known, or can be estimated, at the time.

When attempting to study the weight savings due to the utilization of a lower viscosity fluid, one must consider that, with increasing flow, each fluid will pass from laminar to turbulent flow at a different flow rate. This can be seen by noting that the formulae for Reynolds number include both fluid density and viscosity.

Considering the case where the flow rate of two fluids are increased equally from zero such that both fluid flows start out as laminar, eventually one fluid will break into turbulent and then the other fluid will go turbulent. Since hydraulic lines are of specific sizes (i.e. 3/8, 1/2, 5/8, etc), a system will have several design flow rates for each diameter. For this reason, the study must consider a variable flow rate for the size ratioing.

The plumbing diameter equations for a reference fluid (subscript 1) and a study fluid (subscript 2) for all possible flow conditions are as follows:

Flow Conditions
(reference fluid
to study fluid)

laminar to laminar
$$D_2 = \left(\frac{s_2}{s_1}\,\frac{\nu_2}{\nu_1}\right)^{1/4} D_1 \quad \ldots\ldots\ldots\ldots\ldots \text{(B3)}$$

laminar to turbulent
$$D_2 = \left[\left(\frac{\nu_2}{\nu_1{}^4}\right) Q^3 \left(\frac{3663}{1756}\right)^4 D_1{}^{16} \left(\frac{s_2}{s_1}\right)^4\right]^{1/19} \quad \ldots \text{(B4)}$$

turbulent to laminar
$$D_2 = \left[\left(\frac{1756}{3663}\right)^4 \left(\frac{D_1}{Q^3}\right)^{19} \left(\frac{s_2}{s_1}\right)^4 \frac{\nu_2{}^4}{\nu_1}\right]^{1/16} \quad \ldots\ldots \text{(B5)}$$

turbulent to turbulent
$$D_2 = D_1 \left[\left(\frac{s_2}{s_1}\right)^4 \frac{\nu_2}{\nu_1}\right]^{1/19} \quad \ldots\ldots\ldots\ldots \text{(B6)}$$

When either the fluid viscosity and/or density is reduced significantly, the calculated equivalent tube diameter to attain the same pressure loss per unit of line length is reduced. With the same flow rate in the smaller diameter tube, the fluid velocity will be greater. However, care must be taken that velocities do not exceed values which will cause the pressure rise resulting from abrupt valve closure to exceed the 35% limit specified in MIL-H-5440.

In order to determine the magnitude of weight reduction which could be realized with a lower viscosity CTFE fluid, a study of the Air Force/ Boeing YC-14 Advanced Medium STOL Transport (AMST) aircraft was made. Both the increase in line weight, which would result from the increased tube diameters (and higher fluid density) required to convert the existing hydraulic system from MIL-H-5606 fluid to AO-8 CTFE fluid, and the weight reduction obtainable through use of the 1.8/100 CTFE fluid in lieu of the AO-8 fluid, were estimated.

For that study, the following parameters were used to calculate the fluid velocities which would cause a 35% pressure rise (1,050 psi in a 3,000-psi system) due to abrupt valve closure. Note that the assumed fluid/system compliance values are lower than the measured bulk modulus values in order to account for the typical fluid condition with entrained air.

| Parameter | MIL-H-5606 Fluid | AO-8 and 1.8/100 CTFE Fluids |
|---|---|---|
| Density (at Room Temperature) | 0.85 $g/cm^3$ | 1.336 $g/cm^3$ |
| Adiabatic-Tangent Bulk Modulus (R.T. @ 3,000 psi) | 286,000 psi | 245,000 psi |
| Assumed Fluid/System Compliance | 150,000 psi | 125,000 psi |
| Fluid Velocity for 1,050-psi Pressure Rise due to sudden Valve Closure | 25 fps | 18.8 fps |

In order to minimize computation time, an interactive computer program was written to accomplish the necessary calculations. As written for the YC-14 system study, the program incorporates the weight calculations for tube wall thickness for 21Cr-6Ni-9Mn alloy stainless steel pressure lines for a 3,000-psi system (12,000-psi minimum pressure burst requirement), and 6061-T6 aluminum return lines for design pressures from 600 psi to 1,500 psi (1,800-psi and 4,500-psi minimum burst pressure requirement) depending upon tube diameter. The weight figures are for tubing full of fluid (wet weight) and include a 20% allowance (of tubing dry weight) for end fittings and tube nuts and a 10% allowance (of the tubing wet weight) for tube support clamps.

For that airplane, the minimum hydraulic fluid full-flow design temperature is +50F which is representative of military transports and commercial airliners which are allowed warmup periods whenever they are cold soaked at lower temperatures. Systems for other aircraft, such as all-weather fighters or strategic bombers on ready alert status, must be designed to deliver high flows at considerably lower (subzero) temperatures. For those aircraft the relative system weight penalties to accommodate CTFE fluid will be higher.

## Results

The results of the YC-14 study are shown in the following table. The existing pressure and return line diameters and wall gages are tabulated in the first column. The total lengths of each tube size are tabulated in the second column. The installed wet weights of tubing, fittings, and clamps for each tube size in the existing MIL-H-5606 fluid system are tabulated in the third column. In the fourth column, both the weight ratios for the AO-8 CTFE fluid tubing to MIL-H-5606 fluid tubing, and the installeld weight of tubing, fittings, and clamps for each tube size required for an AO-8 CTFE fluid system are tabulated. In the fifth column, both the weight ratios for the 1.8/100 fluid tubing to MIL-H-5606 fluid tubing, and the installed weight of tubing, fittings, and clamps for each tube size required for a 1.8/100 CTFE fluid system are tabulated.

As seen at the bottom of the tabulations in the third column, the total weight of the MIL-H-5606 fluid plumbing system is 1,538.4 lb. As seen at the bottom of the fourth column, the total weight of an AO-8 CTFE fluid plumbing system would be 3,087.6 lb which represents an increase of 1,549.2 lb (100.7%) over the existing system. As seen at the bottom of the fifth column, the total weight of a 1.8/100 CTFE fluid plumbing system would be 2,207.3 lb which represents an increase of 668.9 lb (43.5%) over the existing (MIL-H-5606 fluid) plumbing system but a decrease of 880.3 lb (28.5%) from an AO-8 CTFE fluid plumbing system.

YC-14 AMST HYDRAULIC TUBING SYSTEM WEIGHTS
FOR MIL-H-5606 FLUID, AO-8 CTFE FLUID, AND 1.8/100 CTFE FLUID

| Tube Diameter and Wall Thickness | Tube Length (feet) | MIL-H-5606 Installed Wet Weight (pounds) | $\frac{W_{AO-8}}{W_{5606}}$ | AO-8 CTFE Installed Wet Weight (pounds) | $\frac{W_{1.8/100}}{W_{5606}}$ | 1.8/100 Installed Wet Weight (pounds) |
|---|---|---|---|---|---|---|
| PRESSURE LINES | | | | | | |
| 3/8 x .020 | 1016 | 134.5 | 1.63 | 219.2 | 1.25 | 168.1 |
| 1/2 x .026 | 494 | 116.9 | 1.72 | 201.1 | 1.42 | 166.0 |
| 5/8 x .033 | 302 | 111.3 | 1.72 | 191.4 | 1.42 | 158.0 |
| 3/4 x .039 | 342 | 182.2 | 1.72 | 313.4 | 1.42 | 258.7 |
| 1 x .052 | 409 | 387.3 | 1.72 | 666.2 | 1.42 | 550.0 |
| | | 932.2 lb. | | 1591.3 lb. | | 1300.8 lb. |
| RETURN LINES | | | | | | |
| 3/8 x .035 | 395 | 34.3 | 1.70 | 58.3 | 1.49 | 51.1 |
| 1/2 x .035 | 426 | 58.4 | 1.65 | 108.0 | 1.60 | 93.4 |
| 5.8 x .035 | 395 | 78.0 | 1.97 | 153.7 | 1.70 | 132.6 |
| 3/4 x .035 | 187 | 50.1 | 2.06 | 103.2 | 1.77 | 88.7 |
| 1 x .035 | 287 | 125.8 | 2.19 | 275.5 | 1.87 | 235.2 |
| 1-1/4 x .035 | 112 | 72.6 | 2.28 | 165.5 | 1.95 | 141.6 |
| 1-1/2 x .035 | 208 | 187.0 | 3.38 | 632.1 | 0.88 | 163.9 |
| | | 606.2 lb. | | 1496.3 lb. | | 906.5 lb. |
| TOTAL WEIGHT | | 1538.4 lb. | | 3087.6 lb. | | 2207.3 lb. |
| Weight Change Relative to 5606 . . . . . . . . | | | | +1549.2 lb. | . . . . | +668.9 lb. |
| | | Weight Change Relative to AO-8 . . . . | | | | -880.3 lb. |